

Engineering Science Series

HEAT ENGINES

ENGINEERING SCIENCE SERIES

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HEAT ENGINES

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PREFACE

Many texts devoted to the general subject covered in this book have been published from time to time under various titles, as Heat Engines, Heat Power Engineering, Applied Thermodynamics, Practical Thermodynamics, etc. The incentive which prompted this effort was the desire for a book containing within reasonable bounds an adequate treatment of the fundamental laws of gases and the laws of thermodynamics together with descriptions of the modern forms of prime movers, their operating characteristics, and representative performance results of each type and size. The author has made a special effort to develop more complete and more satisfying chapters on steam turbines and internal combustion engines than is the case in many of the books now available.

The material was gathered for use in a lecture course for mechanical engineering students at Stanford University. It first appeared in syllabus form and it represents several years of evolutionary development. It is hoped that practicing engineers may also find it worth while as a reference book.

The book contains sufficient material for a thorough three or four unit course extending through the college year. It may be used as the text for a recitation and problem course or as a reference to supplement lecture courses with problems. The book contains many references to original papers and to important articles which have appeared in recent technical periodicals. Thus many of the topics may be considerably amplified if the instructor desires and if the interest of the student makes such collateral reading profitable. On the other hand, by the omission of certain sections which are set in slightly smaller type and which constitute approximately one-third of the text, the essentials may be presented in an abridged course for non-mechanical engineering students.

The author has also endeavored to present in a clear and concise but rigorous manner the fundamental laws of gases and the laws of thermodynamics as announced by the original investigators, with references to their original papers. Among

these investigators I may mention Boyle, Dalton, Carnot, Clausius, Kelvin, Joule, Regnault, Maxwell, Planck, and others. Much credit also belongs to the great engineers and great teachers of engineering, among whom are Tredgold, Galloway, Rankine, Zeuner, Thurston, Cotterill, Peabody, Wood, Ewing, Stodola, and others. I would express reverent acknowledgment to these pioneers and leaders.

Acknowledgment is made to each of the manufacturers and firms who have coöperated generously and substantially in furnishing illustrative material for cuts and performance data of their respective power-plant equipment, and to those authors and publishers who have permitted the use of quotations and reference to material published by them. Appropriate notices of credit are given in footnotes throughout the book. Credit for data and information taken from publications of various scientific and engineering societies and other technical publications has also been duly given in footnotes.

I am indebted to Dr. Stanley C. Herold and Charles M. Cross for constructive criticism of the English and of the subject matter, and for aid in proofreading; and to Dean Theodore J. Hoover for practical aid in typing the manuscript.

Finally, the author wishes to express his gratitude for the kindly and helpful coöperation of the editors, who made many constructive and valuable suggestions.

CHARLES N. CROSS

STANFORD UNIVERSITY
June, 1930

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SYMBOLS

Below is given a fairly complete list of the symbols and their meanings as used throughout this book. Values at special points or specific values are generally indicated by subscripts or prime marks. In many instances, a symbol is used to represent several quantities. Complete and specific information about each usage will be found in the text in the various sections of the book in which they are employed.

- A = Area in sq. ft.; $1/J$; $1/778.57$,
- A_c = air per lb. of carbon,
- A_f = air supplied per lb. of fuel,
- a = areas in sq. ins.; a constant; acceleration in ft. per sec. per sec.,
- B = a constant as specified; Baumé hydrometer reading,
- b = width; some constant as specified,
- B.t.u. = British thermal unit,
- C = temperature in degrees centigrade: constant in equation $PV = C$; element carbon; molecular specific heat,
- c = specific heat in general; a constant; clearance as a decimal,
- c_p = specific heat at constant pressure,
- c_v = specific heat at constant volume,
- $\bar{c}_p$ and $\bar{c}_v$ = mean specific heats for a specific temperature range,
- D. hp. = developed horsepower,
- d = diameter in inches,
- δ = density in pounds per cubic foot,
- Δ = a small finite increment in the value of any variable,
- E = energy in any form,
- e = efficiency in general,
- F = force; factor of evaporation; temperature degrees Fahrenheit,
- f = a correction factor,
- g = acceleration due to gravity (32.174 ft. per sec. per sec.)
- H = total heat (usually in B.t.u. above 32° F.); heat energy used in increasing temperature only; hydrogen,
- h = heat of the liquid of 1 lb. of substance above 32 F.,
- hp. = horsepower,
- I = internal heat,
- in. = inches,

I. hp. = indicated horsepower,

J = mechanical equivalent of one B.t.u.; 778.57 ft.-lbs.,

j_p = specific heat at constant pressure in ft.-lbs.,

j_v = specific heat at constant volume in ft.-lbs.,

K = constants in general; the constant in the equation $PV^n = K$,

KE. = kinetic energy in ft.-lbs.,

k = ratio of specific heats c_p/c_v or $\bar{c}_p/\bar{c}_v$,

kw. = kilowatt,

L = length usually in feet; boiler heat loss,

l = length usually in inches,

M = weight in pounds unless otherwise specified,

m = decimal fraction of a mixture of liquid and vapor which is liquid; molecular weight,

mR = molecular gas constant,

N = entropy in general; revolutions per minute; nitrogen,

n = an exponent; revolutions per minute; specific values of entropy,

P = pressure in general; pressure in pounds per sq. ft.; power,

p = pressure in pounds per sq. in.,

Q = heat quantities in general,

q = a small portion of heat of which Q is the total,

R = gas constant; relative velocity in ft. per sec.,

r = ratio of expansion; latent heat of evaporation; rotation loss,

r.p.m. = revolutions per minute,

ρ = internal energy in B.t.u. in one lb. of dry saturated steam,

S = speed in general; absolute velocity; sulphur,

s = seconds of time; piston speed in ft. per minute,

T = absolute temperature (in degrees F. unless otherwise specified),

t = thermometer readings (usually degrees F.),

U = higher heating value of fuels in B.t.u. per lb.,

UE. = unit of evaporation,

u = lower heating value of fuels in B.t.u. per lb.,

V = volume of M lbs. of a gas or vapor in cu. ft.,

v = volume of one lb. of a gas or vapor in cu. ft.,

W = work in general or external work both in ft.-lbs.,

x = quality of wet steam, a decimal number,

y = ratio of pressures; total rotation loss as a decimal fraction of E .

HEAT ENGINES

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CHAPTER I

POWER FROM NATURE'S FORCES

SYSTEMS OF UNITS

1. **Introduction.** One of the most important functions the engineer performs is to transform nature's stores of energy into useful forms. The demand for more power to carry on the world's work is increasing at an accelerating rate. The essentials and luxuries of our civilization could not be produced in the vast quantities required, without the aid of mechanical and electrical power transformed from waterfalls and fuels in ever increasing amounts. The transformation and application of inanimate forms of energy on a titanic scale in carrying on the work of mankind constitutes the outstanding characteristic which distinguishes modern civilization from preceding ones.

The greatness of Rome was built up and maintained by the muscular power of vast numbers of human slaves. Comparatively, only a few favored individuals were able to enjoy the benefits and privileges of that greatness. But now the masses are, in a large degree, relieved from the laborious toil of those early times which continued down to the invention of the steam engine about 1700 A.D. It is only the large-scale transformation and application of nature's stores of energy to man's purposes that have made possible a generous per capita production and distribution of the desirable commodities of life. The discoveries and inventions of the scientists and engineers have been the prime factors in producing the modern developments. The chief and almost universal benefit to humanity is *freedom from the slavery of muscular toil*. There have been, down through the ages, great philosophers, great law givers, great warriors, and great statesmen. Some of the greatest flourished in the earliest

times of recorded history. But not one of them, nor all of them combined, were able to produce a code of ethics, a system of laws, or a plan of social organization which was not founded upon slavery in some form for a large percentage of the race. To the scientist and to the engineer must be given a large measure of credit for harnessing nature's forces, thus abolishing slavery. It is because production and distribution can be of such great magnitude that the masses of the present period in history are enjoying many conveniences and even luxuries which kings of earlier days could not command. The differences in the styles of living between the highest and the lowest of the land have been greatly reduced because the inanimate forces of nature have been put to work. The end of progress in this direction is not in sight. Indeed, the indications are that the developments within the next seventy-five or one hundred years will far exceed in magnitude and importance everything which has yet been done in converting nature's forces to the uses of man.

2. Sources of Power. There are at present three natural sources of power which are sufficiently developed to be worthy of mention. They are, in a descending order of importance, fuels, waterfalls, and winds. The last-mentioned resource is extremely variable from day to day and from place to place. It is by far the least important one of the group. At the present time, more than ninety-eight per cent of the world's power is obtained from the first two groups. It has recently been estimated ¹ that the total commercially profitable water-power resources of the United States amount to about fifty-four millions of horsepower. Probably about twelve millions of horsepower are developed at the present time.² It is also estimated that, for the year 1928, approximately seventy-five millions of horsepower capacity were in use in power plants, large and small, within the United States. The entire available water power of this country, if fully developed, is insufficient to meet the power demand at the present time. The demand is growing very rapidly. Ultimately, water

¹ See the *Transactions of the First World Power Conference*, Vol. 1, 1924.

² *U. S. Geol. Survey Report on Prime Movers*, 1927; *Power Plant Engineering*, Mar. 15, 1927, p. 386.

power will be sufficient to supply only a small fraction of the power load. Hence, a long look into the future indicates that fuel must continue, as now, to be the most important source of power. As long as the present general climatic conditions prevail upon the earth, the water power resources will be perpetuated. Unless some unexpected cataclysm arrives, this should continue for some millions of years.

3. Fuel Resources. Geological investigations indicate conclusively that, as in the case of all other earthly stores, the supply of fuels in the form of coal, petroleum, and natural gas is strictly limited. Furthermore, when these stores are exhausted, it is very unlikely that they will ever be renewed. Recent estimates, based upon the latest and most authoritative information as to fuel resources and future rates of consumption, indicate that the known fuel stores will probably become exhausted within the next two hundred or three hundred years. This is, indeed, a very short time in the span of recorded history, and only an infinitesimal moment compared to the millions of years since geological evolution began. Our captains of industry and our engineers have a real and an immediate problem to solve in utilizing and conserving to the highest possible degree the earth's limited store of fuels.

When the fuel stores become inadequate or approach exhaustion, as they surely must, there is at least a partial solution at hand. The torrid zone is peopled largely by the dark races, which have, as yet, contributed comparatively little to the welfare and advancement of mankind. Also, in the torrid zone, where there is an abundant rainfall, vegetable growth is very rank and rapid, due to the large amount of heat energy received from the sun throughout the year. The large potential man power of the dark races may be put to work raising rapidly growing plant life which may be most suitable for the production of alcohol. This large-scale production of alcohol, would, perhaps, go a long way toward supplying the world demand for power which cannot be met by the earth's waterfalls. This supply of fuel from vegetable growth would be inexhaustible so long as our sun continues to radiate heat and light at anywhere near its present rate.

There is also another possible solution. Certain scientific developments from the study of radium, the electron, and the structure of matter seem to indicate that it may be possible to unlock and transform for man's purposes the vast stores of energy resident within the atom. If this should come to pass, the probabilities are that the quantity of energy which would become available per unit weight of matter would be hundreds or even thousands of times greater than now obtained from the combustion of fuels. Besides the materials which we now consider to be ideal fuels might be displaced by other materials more satisfactory for atomic transformations. This new and vast store of energy may become available long before the supply of fuels is exhausted. However, too little is known, as yet, about atomic energy and its transformations to permit us to more than speculate about it. But intensive research is being carried on by scientists and engineers with increasing promise of surprisingly successful results in the not-distant future. Meanwhile it will continue to be an important duty of the engineers to conserve, in the highest possible degree, the fuel resources of the earth.

4. Thermodynamics. The subject of thermodynamics in the broadest sense is that branch of physico-mathematical science which deals with transformations of heat energy into other forms; such as mechanical work, electrical energy, and chemical energy. From the point of view of the engineer, thermodynamics deals specifically with transformations of heat energy into useful mechanical work. By its mode of dealing with certain fundamental physical laws, mathematical relations connecting the variables which are involved in transformations of energy are developed. By its processes, the thermal properties of substances are studied, and the ones best suited to serve as media for carrying on transformations of energy in the generation of power are selected. By thermodynamic methods of reasoning, the ideal processes by means of which the maximum mechanical effect may be obtained from a given quantity of heat energy have been determined. The maximum possible ratio of mechanical work to the heat energy supplied can be determined by means of

thermodynamic equations. By thermodynamic processes, mathematical expressions which give performance results of ideal and actual prime movers operating under given conditions are readily found.

5. Heat Engines. A heat engine is a device or a combination of devices, by means of whose operation, heat energy is transformed into useful mechanical work. The engine may be considered in two ways. First, consider *an ideal engine* operating in an ideal manner to give the highest theoretical performance possible. Second, we may discuss *a practical engine*, which, operating in a practical manner, will give the largest economic result practicable for a given investment of energy, materials, and labor. In studying and developing either type, the fundamental processes and laws of thermodynamics must be known quantitatively, and they must be applied skillfully if heat energy is to be transformed most advantageously into work. The practical side of the subject of heat engines includes a detailed study of devices and expedients employed in obtaining the highest possible efficiency for transforming energy. Hence, a large part of the subject of heat engines is devoted to the discussion of *efficiency*. The term *efficiency* represents an abstract ratio. In general terms, *it is the result accomplished divided by the effort made*. The *efficiency of a heat engine* is the ratio of the useful work done to the heat energy supplied to the engine. Both quantities of energy must be expressed in the same kinds of units. In general, efficiency will be designated by the symbol e . In the form of an equation, one may write

$$e = \frac{\text{Useful work}}{\text{Energy supplied}}.$$

When the quantities are all expressed in heat units, e is called *thermal efficiency*.

In all practical engines, useful work is performed by virtue of the change of volume of a substance which takes place when heat energy is imparted to it. The substances universally used are always in the gaseous state. The reason for this will become apparent as the subject is developed in the subsequent chapters.

The *first heat engine* of which there is any record was probably invented and described by the Greek mathematician, Hero, about 130 B.C. Hero's engine has now been developed into the much-used reaction steam turbine. Hero probably built and operated his engine as a scientific curiosity, but he never put it to practical use. The first heat engine actually to perform useful work in appreciable amounts was invented in 1697 by Thomas Savery. The Savery "*Fire-engine*" was used to pump water from the English coal and tin mines when they became so deep that pumping could no longer be done economically by use of horses.¹ This type of engine, in a modified form, is still used where a portable and cheap pump is desired. Contractors use it for pumping dirty and gritty water from excavations. It is now known as a *pulsometer*. In the Savery pump, the driving steam comes into direct contact with the cold water, which acts as a piston. Consequently, a large part of the steam is condensed, and the steam consumed per unit of work done is very large. This disadvantage, however, is compensated for by the cheapness, portability, and simplicity of this type of pump.

The fundamental reason why heat energy can be made to do useful mechanical work was not understood until about 1800. Preceding that date, the old *phlogiston* theory of Stahl and Becker prevented much progress. The phlogiston theory considered heat to be an elemental substance which separated from combustibles when they burned. However, the phlogiston theory served a useful purpose as it was a stepping stone to something better. The modern scientific era had its beginning about 1800. The researches and publications of Lavoisier, Count Rumford, Sir Humphrey Davy, and others of that day placed science on a solid foundation which has made possible the wonderful scientific and engineering developments of modern times. Among others, the subject of thermodynamics and heat engines has been evolved during the comparatively short span of 125 years. In a historical sense, such a period of time for a development which has affected the welfare of the human race so profoundly is very short.

¹ See Elijah Galloway, *History of the Steam Engine*, 1826, for an illustrated description of Savery's Fire-engine.

6. Fundamental Units. In order to investigate and measure, in a quantitative manner, the materials and forces of nature, it is necessary to use some standards of measurement by means of which comparisons may be made. Nature has not directly provided any convenient standards which are absolutely unchanging and easily reproduced. Hence it has been necessary for man arbitrarily to adopt artificial standards called *units of measurement*. The earliest investigators chose many kinds of units which, as time passed and knowledge increased, turned out to be inconvenient, unrelated, and unsuited for the purposes intended. Consequently much confusion in comparing results of investigators has resulted and still persists. As science has developed and the laws of physics, chemistry, and mechanics have been determined, it turns out, strangely enough, that only *three kinds of fundamental units are necessary*. All of Nature's laws concerning materials and forces may be expressed quantitatively in terms of these three fundamental units, or in terms of units derived from them.

In engineering mechanics, the three units considered as fundamental are those of *length, time, and force*. The physicist considers the three fundamental units to be *length, time, and mass*. In either case, all other units are derived from these three by definition and mathematical relations. The value of each fundamental unit is chosen arbitrarily and becomes accepted by common agreement among engineers and scientists. Each selection of the values of this trinity gives rise to a *system of units*.

7. Fundamental Units Defined.¹ Engineers use as their unit of length the *English foot*. It is defined by law in the United States as 1200/3937 of the length of the International Meter of Paris.

The unit of force is the *pound*. It is the force required to sustain (in vacuum) a standard pound body ($1/2.2046$ of the International kilogram body of Paris) against the pull of the earth's gravity at the standard locality. The standard locality is at sea level and 45 degrees of latitude, where the acceleration

¹ See L. A. Fischer, "History of Standard Weights and Measures of the United States," *Bureau of Standards Bulletin No. 3, 1905*.

due to gravity is 32.1740 feet per second per second. This equals 980.665 centimeters per second per second. This is the unit of force in the British Gravitational System of Units.

It is unfortunate that the word *pound* is used to indicate both a *force* and the *mass* of a body. This gives rise to considerable confusion at times.

The *poundal* is that force which, when applied to the pound body for one second, would give it an acceleration of one foot per second per second. Its value is $1/32.174$ of the pound force. This is the unit of force in the British Absolute System of Units. It is seldom used.

TABLE I
SYSTEMS OF UNITS ¹

NAME OF SYSTEM	FUNDAMENTAL UNITS		
	Length	Force	Time
British absolute (used but little)	Foot	Poundal	Second
British gravitational (foot-pound-second)	Foot	Pound	Second
Metric absolute (c.g.s. system)	Centimeter	Dyne	Second
Metric gravitational (kilogram-meter-second, used but little)	Meter	Kilogram	Second

The *kilogram* is the force required to sustain the International kilogram body of Paris (in vacuum) against standard gravity as defined above. It is the unit of force in the Metric Gravitational System of Units. As in the case of the word *pound*, the word *kilogram* is ambiguous; its use to denote a unit of force is unfortunate.

¹ For all derived units see any text on mechanics, or Mark's *Mechanical Engineers' Handbook*.

The *dyne* is the force which when applied to the standard gram body for one second would produce an acceleration of one centimeter per second per second.

The *second* is the unit of time used in all systems. It is $1/86400$ part of a mean solar day.

The use of these units give rise to four systems as shown in Table 1.

CHAPTER II

ENERGY AND MATTER

8. Definition of Technical Terms. A definition of any term is always an attempt to explain it in simple language which gives as complete an understanding of it as possible. Defining a term may be compared to the process of reducing a complex fraction to its simplest form. Our experiences in life have brought to our attention a number of concepts which seem difficult to resolve into simpler components. The phenomena of time, space, and energy belong to this group of elementary concepts observed in nature.

9. Energy. The term energy has been given to one of the elementary phenomena of nature. Many attempts have been made to define it. The definition most frequently used was formulated by J. Clerk Maxwell as follows: "*Energy is capacity to do work, that which in all natural phenomena is continually passing from one portion of matter to another.*"

There are two broad classifications of energy and many forms common to both classes. When a motor car moves along the highway, it possesses a definite store of energy by virtue of its weight and speed. If the car is brought to rest, this energy of motion is used up in internal friction at the brakes, or in carrying it up an incline. If a car having a certain speed is started up a uniform grade, and allowed to coast to rest at the top of the incline; all the energy of motion it possessed at the foot of the incline has been used up in carrying the car to the top against the force of gravity. (For simplicity, it is assumed in this illustration that all minor energy losses and uses, including windage, friction, and battery charging, are zero). If the car is then permitted to coast down the incline under the influence of gravity alone, it will regain in reversed sequence exactly the quantity of energy it lost on its ascent.

While the car was in motion ascending or descending the grade, it possessed different amounts of *energy of motion* from instant to instant by virtue of its instantaneous velocity. Energy of motion is called *kinetic energy*. When the car was at rest at the top of the incline it possessed no kinetic energy. But because of its position, it was capable of acquiring velocity and kinetic energy simply *by virtue of its position* in the earth's field of gravitation. In this condition it is said to possess *potential energy*.

There are many forms in which energy is manifested. These are readily distinguished and may be classified in several ways into primary, secondary, and tertiary forms. Which ones are to be considered primary, which secondary, and which tertiary, depends partly upon the previous educational training and experience of the classifier. The chemist's classification would probably differ in some details from that of the physicist, while the engineer's would differ from the classifications of both. The following tabulation is one of several which might be made.

TABLE 2
CLASSIFICATION OF ENERGY

<i>Potential</i>	Electronic	{	Electrical	{	Chemical reactions including combustion of fuels
		{	Chemical		
<i>Energy</i>	Atomic	{	Chemical	{	Chemical reactions including combustion of fuels
	Molecular or Heat				
	Solar	{	Radiations		
	Lunar	{	Tides		
	Stellar		Waterfalls		
		{	Winds		
<i>Kinetic</i>	Psychic	{	Human	{	
			Lower animals		
	Mechanical	{	Animals		
	Machines				

Any classification of the forms of energy, such as that shown in Table 2, cannot be permanent. Research and investigation are continually bringing forth new information as to the nature and inwardness of matter, energy, and life. Until all the important secrets of nature have been discovered, the final and permanent classification cannot be written. Ultimately it may turn out that electrical, chemical, and psychic forms of energy are identical. For the time being, however, the arrangement shown in Table 2 will probably serve to show the relationships of the forms of energy as well as any other that could be formulated in the light of present knowledge.

10. Nature of Matter and Energy.¹ The intensive research which has been carried on during the past fifty years into the fundamental constitution of matter indicates rather conclusively that the universe of our acquaintance is altogether made up of *positive* and *negative charges of electricity*. This evidence has been gathered and is being added to very rapidly by such scientists as Professor Joseph J. Thomson, Madame Marie Curie, Professor Hendrick A. Lorentz, Sir Ernest Rutherford, Professors Robert A. Millikan, and Albert A. Michelson, Niels Bohr, Albert Einstein, Irving Langmuir, Arthur H. Compton, and others.

The inner nature of the electric charge is unknown. But all the masses of the universe seem to be made up entirely and only of great aggregations of countless trillions of electric charges. The properties of the subdivisions of matter which we call elements seem to be due solely to the number and arrangement of positive and negative charges in clusters, which are called atoms. Sir Ernest Rutherford has given the name, *proton*, to the positive charge, which has been isolated and identified within the past ten years. In 1895, Professor J. J. Thomson proved that the cathode ray of a Crookes tube with which he had been working for a number of years is a stream of negatively charged particles moving at very high velocities. He named these charges *electrons*. The two charges are of opposite sign and

¹ See James A. Crowther, *Molecular Physics*.

John Mills, *Within the Atom*.

Robert A. Millikan, *The Electron*.

exactly balance or neutralize each other electrically when brought together.

Measurements, checked repeatedly, indicate that the diameter of the electron is approximately 2000 times that of the proton. On the other hand, the mass of the proton is about 1850 times the mass of the electron.

Mass, energy, and force all seem to be manifestations due to the movements or to changes in the velocity of movement of the electrons and protons. A clear picture as to just how and why this may be so is not yet available. Information upon these matters lies on the outermost border of the fields of scientific exploration and there is still much to be learned. Explanations for many phenomena are as yet but little better than speculations. But if the evidence in hand endures the tests of time, it means, in the final analysis, that *energy is matter* and *matter is energy*. Undoubtedly, continued research will show that there are certain qualifications and limiting conditions which will have to be applied to the sweeping statement that energy is matter. Transformations from one to the other are evidently taking place in nature at all times under special conditions and limitations scarcely understood as yet. These ideas are so revolutionary, in view of the historical background, that they appear unacceptable without extensive qualification.

The first thought is that if mass is not a constant under all conditions, then the *Newtonian mechanics* must be all wrong. The explanation is that Newton's laws are correct as far as they go, but they are in reality only first approximations. They correspond to the first two or three terms in a long mathematical equation in which each succeeding term becomes of less importance. At very high velocities approaching that of light, the higher terms developed from the theory of relativity must be taken into account in order to obtain precise results. The velocities of moving bodies with which the engineer deals are so small compared to the velocity of light that the formulas of the Newtonian mechanics give results sufficiently precise for all practical purposes. But for the scientist, dealing with problems of the electron or celestial mechanics, the above statement no longer holds.

11. Kinetic Theory of Heat.¹ In a study of the subject of heat engines, we are interested only in that kind of phenomena known as heat, its nature, its manifestations, and the transformations necessary to obtain useful work from it. According to the kinetic theory of gases, which is now universally accepted among scientists and engineers, heat energy is kinetic energy, largely due to the confused oscillatory vibrations of molecules. In the case of fluids there are movements of translation of the molecules in straight lines as well as rotational and vibratory motions. Rudolf Clausius computed the average velocities of the molecules of several gases for the condition of atmospheric pressure and temperature of 32° F. to be as follows:¹

Oxygen.....	1514 feet per second
Nitrogen.....	1616 feet per second
Hydrogen.....	6050 feet per second

The fastest-moving molecules in each gas are probably traveling four or five times as fast as the average velocity. The slowest-moving molecules may not be moving more than 1/100 as fast. The kinetic theory is beautifully developed in the references given. Modern research into the structure of matter and the nature of energy has served to confirm and add to the kinetic theory, until to-day it is more firmly established than ever.

12. Conservation of Energy. The development of the idea that all forms of energy can be transformed from one to another upon an equivalent basis was a slow evolution. Many scientists contributed to the development of the Law of Conservation of Energy. Professor Peter G. Tait states² that "Newton had completely enunciated the *conservation of energy* in ordinary mechanics, Davy's experiments brought heat under the same law, and that, therefore, in the beginning of the present century the Dynamical Theory of Heat was completely *established* although not *developed*. The development has been furnished

¹ See James Clerk Maxwell, *Theory of Heat*.

John Tyndall, *Heat a Mode of Motion*.

Thomas Preston, revised by J. R. Cotter, *Theory of Heat*.

Encyclopaedia Britannica, 11th ed., vol. 18, p. 657.

Leonard B. Loeb, *Kinetic Theory of Gases*.

² Philosophical Magazine, vol. 28, 4th series, 1864, p. 289.

since by the experiments and reasonings of Joule on Electric Thermo-Dynamics, his experimental determinations of the mechanical equivalent . . . ; and by the theoretical writings of Helmholtz, Clausius, Rankine, and Thomson."

The *Encyclopaedia Britannica*¹ states that one of the earliest statements of the principle of the conservation of energy was made by Karl Friedrich Mohr, a German pharmacist, in a paper, *Über die Natur der Wärme*, published in *Zeitschrift für Physik* in 1837. He gave one of the earliest general statements on the doctrine of conservation in these words, "Besides the 54 known chemical elements there is in the physical world one agent only, and this is called Kraft (energy). It may appear, according to circumstances, as motion, chemical affinity, cohesion, electricity, light, and magnetism, and from any one of those forms it can be transformed into any of the others."

The question as to whom credit belongs for priority in the discovery, formulation, and publication of the conservation law was debated hotly and even bitterly in European scientific circles for more than three decades. Peter Tait was inclined to give the credit to M. Sequin for his attempt to use computations of the mechanical equivalent of heat from the values of the two specific heats of gases as proof of the law. His results and deductions were published in 1839. Tyndall after much study of the question of priority finally gave credit to Julius R. Mayer, a physician of Heilbronn in Germany, for being the first to publish in 1842 an adequate and acceptable exposition on the mechanical equivalent of heat.² The results published at that time, however, lacked experimental confirmation. Tyndall gives to James Prescott Joule complete credit for furnishing the first authentic experimental proof of the law. Joule was occupied from 1840 to 1849 in carrying on and checking completely his experiments to determine the value of the mechanical equivalent of heat.

It is manifestly unfair to give the entire credit for the discovery of the law of conservation to the two men, Mayer and Joule.

¹ *Op. cit.*, 14th ed., vol. 15, p. 660.

² See Liebig's *Annalen*, vol. 42, p. 233.

Tyndall, *Philosophical Magazine*, 1863-64.

Tait, *Philosophical Magazine*, 1864, Pt. 2, p. 289.

Many able scientific men preceding them had felt that some such law existed, since they had noted applications of it to certain groups of phenomena which came under their respective observation. Galileo, Rumford, Davy, Bernoulli, Lavoisier, Carnot, Classius, and others were among them. But their scientific caution and conservatism prevented them from publishing it broadcast without more complete experimental evidence to confirm it. Before Carnot died in 1832, he was firmly convinced of the dynamic theory of heat involving the idea of the equivalence of heat and work. He left in his notes detailed plans for several experiments to prove the theory. Joule, later and independently, devised and performed almost exactly the same experiments as those outlined by Carnot.

The law of the conservation of energy has been stated in various ways as follows:

1. *Energy is indestructible.*
2. *The quantity of energy in the universe is constant.*
3. *Energy of whatever form required to change a system of bodies from one state to another is independent of the particular intermediate states through which it passes.*

This law, as in the case of all of Nature's great fundamental laws, is not subject to mathematical proof, but it has never yet been found to be inconsistent when all the facts about any phenomena were known. It is the fundamental basis of all physical, chemical, electrical, and astronomical reasoning, as well as the basis of all energy transformations.

In the light of all the recently acquired knowledge regarding electrons and atoms, and their structure and behavior; it would seem that a more comprehensive statement of the law of conservation may be necessary in order to make it perfectly general. In that case the law should then be stated as follows:

1. *Energy and matter are indestructible.*
2. *The sum of the total quantity of energy and matter in the universe is a constant.*
3. *Energy and matter, each in a number of forms are mutually convertible from one to the other upon some equivalent and constant basis.*

13. Perpetual Motion Machine of the First Class. A machine or device which might possibly be invented to operate forever, giving out a continuous supply of useful energy without transforming other energy or matter in any form, would be a perpetual motion machine of the first class. It would evidently be creating energy out of nothing. So long as the law of conservation holds, it will be an impossibility to invent or operate such a machine. The statement that a perpetual motion machine of the first class is an impossibility is only a revised statement of the conservation law.

CHAPTER III

HEAT AND WORK

14. Work. Mechanical work equals the product of force and the distance through which it operates. In symbols, it becomes

$$(1) \quad W = F \times D.$$

When F is measured in pounds and D in feet, the product is called *foot-pounds* of work; if in kilograms and meters, it is called *kilogram-meters*. The expression *inch-pounds* is also used.

15. Kinetic Energy. Kinetic energy also is work. It is measured in foot-pounds, inch-pounds, or kilogram-meters, as the case may require. According to the Newtonian mechanics

$$(2) \quad KE = \frac{MS^2}{2g}.$$

In equation (2)

KE = kinetic energy,

M = weight of substance moving,

g = acceleration due to gravitational field,

S = uniform velocity per second.

When M is measured in pounds, S in feet per second, and g is taken as 32.17, the unit of KE is in foot-pounds, as is W in equation (1).

If F is a variable, equation (1) becomes

$$(3) \quad dW = f(F)dD,$$

and

$$(4) \quad W = \int_{D_1}^{D_2} f(F)dD.$$

16. Power. Power is the rate of doing work. It involves besides the force and distance, the time factor. In symbols, it becomes

$$(5) \quad P = \frac{FD}{\text{time}}.$$

Horsepower (symbolized hp.) is one particular rate of doing work. It is used as the unit of power in all English-speaking countries.

One *horsepower* is developed when work is done at the rate of 33,000 foot-pounds per minute; or 550 foot-pounds per second. In symbols, it becomes

$$(6) \quad hp. = \frac{FD_m}{33\,000} = \frac{FD_s}{550}$$

where

F = force in pounds,

D_m = distance in feet per minute,

D_s = distance in feet per second,

kw. = kilowatt.

By the U. S. Bureau of Standards ¹

$$(7) \quad 1 \text{ kw.} = 1.3405 \text{ hp.}$$

17. Man's Sources of Power. Several of the items listed in Table 3 are unimportant, and probably will always remain so.

TABLE 3

MAN'S SOURCES OF POWER

SUN AND MOON	{ Light and Heat Gravitational force	{ Life Waterfalls Wind and waves Tides Waterfalls	{ Animal Plant { Fuels
EARTH	{ Heat Matter	{ Geysers Volcanoes Chemical reactions	{ Heat Electricity

The possibilities of obtaining appreciable quantities of power from waves and tides, on a commercially profitable basis, are very remote. Geysers and volcanoes, as a source of power, have not been thoroughly investigated, but moves in that direction

¹ See Circular of the U. S. B. S., No. 34, 1912.

are beginning to be made.¹ For the practical purposes of the engineer there are, at the present time, two very important sources of energy from which power may be converted to the use of mankind. The first is *combustion of fuel*; the second is *falling water*. The fuel resources are, in the final analysis, stored light and heat energy received mostly from the sun during past geological ages. Waterfall is due to the heat energy received from the sun at the present time and to the fact that the water exists in a gravitational field.

From the beginning of man's existence down to the advent of the age of mechanical power with the invention of the steam engine by Thomas Savery, about 1700, men and animals had been the only agency used in performing the work of mankind. Savery was the first to employ the term *horsepower* as a unit to measure the performance of the steam engine.² In 1784 Watt carried out a series of experiments, measuring the capacity of men and other animals to do work. He found that the large English draft horse working for a few minutes at a time was able to do work at the rate of about 33 000 foot-pounds per minute. Later Coulomb, Navier, Poncelet, and others conducted similar experiments. Professor William J. M. Rankine collected all this data and published the most reliable of the results. Tables 4 and 5 are extracted from Rankine's work.³

TABLE 4
WORK OF A MAN AGAINST KNOWN RESISTANCES

KIND OF WORK	FORCE IN LBS.	VELOC- ITY F.S.	HRS. DAY AT WORK	FT. LBS. PER SEC.	HP.
Climbing ladder (own weight only)	143	0.5	8	71.5	0.13
Pushing or pulling capstan or oar	26.5	2.0	8	53.0	0.0965
Turning a crank	18.0	2.5	8	45	0.082

¹ See Mechanical Engineering, Dec., 1925, pp. 1175-1178.

² See Samuel Smiles, *Lives of the Engineers*, vol. 14.

³ See William J. M. Rankine, *The Steam Engine*, pp. 84-90.

All things considered, a man seems to be most efficient for continuous service when he works one-third of the 24 hours in a day at one-third of his maximum speed, and exerts one-third of his maximum force producing about one-tenth of a horsepower. Hence to generate 5000 horsepower for eight hours per day would require 50 000 men. If a floor space two feet by four feet be allotted to each man, 400 000 square feet would be necessary. This would require forty stories in a building one hundred feet square, or over nine acres of ground.

TABLE 5
WORK DONE BY HORSES ¹

KIND OF WORK	FORCE LBS.	VELOC- ITY F.S.	HRS. DAY	FT. LBS. PER SEC.	HP.
Thoroughbred trotting drawing light railway carriage	30.5	14.7	4	447.5	0.812
Draught-horse drawing boat or cart at a walk	120	3.6	8	432	0.785
Draught-horse in a treadmill	100	3.0	8	300	0.55

A horsepower year (8760 hours) would cost, at \$3.00 per day of eight hours per man \$32 400. By good water power, such as the Pitt River developments of the Pacific Gas and Electric Co., or the Niagara Falls Power Company's plant, a horsepower year can be had at a cost of from \$20 to \$30. It is thus seen that man power, even if it were obtained for ten to thirty cents per day, as in China, is extremely expensive power. That statement is also true in large degree for all other animal power.

If the figures of Table 6 are nearly correct, about 7 horsepower is available for each unit of the population. This installed capacity is undoubtedly capable of generating power continuously twelve hours of the twenty-four. At the rate of one-

¹ Other animals are capable of developing power approximately as follows:

Large ox.....	0.36 hp.	Velocity two-thirds of horse.
Mule.....	0.30 "	Best velocity same as horse.
Burro.....	0.10 "	" " " " " "

tenth of a horsepower per man for eight hours, about 100 slaves similar to the galley slave of ancient Rome would be required for each of the 120 millions of people in the United States in order to produce the necessities and luxuries now made available by the utilization of all this power from nature's forces. This would be a total of about twelve and one-half billion men slaves. This is probably about five times the entire number of people living on the earth.

TABLE 6

ESTIMATED CAPACITY FOR GENERATION OF POWER IN THE UNITED STATES
IN 1928

SOURCES	MILLIONS OF HORSE- POWER	SOURCES	MILLIONS OF HORSE- POWER
Horses and mules	10.0	Marine power	5.0
Tractors	100.0	Mining industry	4.0
Automobiles	500.0	Electric railway	5.0
Steam railroads	150.0	Isolated electric plants	5.0
Manufacturies	30.0	Mills flour, saw mills and irrigation	3.0
Central power stations	38.0	Total	850.0

18. Relation between Heat and Work. In 1789, Count Rumford found that heat may be produced from mechanical work without the agency of fire. *He found its amount to be proportional to the quantity of mechanical work expended.* Rumford made this observation while superintending the manufacture of cannon in Munich for the British Army. In 1799, Sir Humphrey Davy performed his classical experiment of melting ice by rubbing two blocks of it together vigorously. *He found that the quantity of ice melted is proportional to the amount of work expended in rubbing.* Davy concluded that the melting was due to increased activity of the elemental units of matter.

19. Joule and the Mechanical Equivalent of Heat. James Prescott Joule (1818-1889), English physicist, was largely a self-educated man. However, he studied for a time under John

Dalton. Joule was one of the first to appreciate the great importance of accurate and precise physical measurements. Until Joule's time, few investigators had given to this detail the study and attention it deserved. His careful, painstaking efforts to do accurate quantitative work constitutes his greatest achievement.

His accidental meeting with Professor William Thomson (Lord Kelvin) in 1847 and the continued association of these two men had a profound effect upon scientific progress. The mental endowments of the two men seemed to be sympathetically complementary to an unusual degree. Lord Kelvin was equipped with one of the greatest intellects, capable of dealing with natural phenomena on an abstruse mathematical basis. Joule was peculiarly fitted to carry out successfully the precise laboratory measurements necessary to supplement Kelvin's mathematical abstractions, and thus establish their validity.

In 1843, Joule published ¹ for the first time the results of several years' work on determinations of the value of the mechanical equivalent of heat. He published the results of eight different determinations varying from 487 to 1040 foot-pounds as the work required to raise the temperature of one pound of water one degree Fahrenheit. The average of all the results was 882 foot-pounds. Such discordant results did not make the impression upon the scientific world which this important subject deserved. By persistent effort and repeated trials his skill increased resulting in better agreement among his different results. In 1849, he announced that the mean value of more than one hundred trials was 772 foot-pounds. This value stood unchanged for about thirty years. Henry A. Rowland,² physicist at Johns Hopkins University, in 1878, found by more refined methods the value **778 foot-pounds**. Since Joule's time, many other investigators ³ have made determinations of the value of the mechanical equivalent. Their results nearly all fall between 777 and 778. An exhaustive research on the properties of steam has recently been made by a committee of the American Society of Mechanical

¹ Philosophical Magazine, vol. 23, 1843, p. 435.

² Proceedings of the American Academy, vol. 15, 1880, p. 75.

³ See Reynolds and Moorbey. Philosophical Transactions, vol. 190A, 1898, p. 301. Also Barnes, Philosophical Transactions, vol. 199A, 1902, p. 149.

Engineers. As a part of the research the mechanical equivalent of heat has been redetermined by the United States Bureau of Standards. The estimated error of the new value is one part in 5000.

The value announced is *one mean gram calorie equals 4.189 Joules*.⁴

Since

$$1 \text{ B.t.u.} = 453.592 \times 5/9 = 251.995 \text{ calories,}$$

or

$$1 \text{ B.t.u.} = 251.995 \times 4.189 = 1055.62 \text{ Joules,}$$

and

$$1 \text{ B.t.u.} = \frac{1055.62 \times 10^7 \times .3937}{980.665 \times 453.592 \times 12} = 778.57 \text{ ft.-lbs.}$$

The value of the mechanical equivalent varies slightly with the pressure of the water and inversely with the value of the acceleration due to gravity. For all practical purposes the value 778 is sufficiently accurate.

It should be recalled that the Joule is 10^7 ergs. One erg is a dyne-centimeter. Hence, the Bureau of Standards value of the mechanical equivalent is based upon the c. g. s. system of units.

The investigations of Joule furnished the foundations upon which the dynamical theory of heat was finally and firmly established. His name will always occupy the foremost place among physical philosophers. As a memorial to him the mechanical equivalent has been named *Joule's equivalent*. It is universally represented by the letter J.

20. First Law of Thermodynamics. The first law of thermodynamics is, perhaps, a restricted statement of the law of conservation, namely: heat and work are mutually convertible from one to the other in a definitely fixed ratio. The first law may be expressed more concisely by an equation as follows:

$$(8) \qquad W = JQ,$$

⁴ Mechanical Engineering, Feb., 1928, p. 153, and Feb., 1929, p. 109.

(A mean calorie is 1/100 of the heat required to raise the temperature of one gram of distilled water from 0° C. to 100° C. at atmospheric pressure).

in which

W = foot-pounds of mechanical work,

Q = quantity of heat in British thermal units,

J = Joule's equivalent = 778.57 ft.-lbs.

21. Degradation of Energy. The conservation law indicates that it might be possible for any form of energy to be converted into any other form. This is undoubtedly true if the necessary conditions are provided. However it does not state that certain transformations are naturally and easily accomplished, in fact are continually taking place under the normal conditions provided by nature. Neither does the conservation law state that certain kinds of energy may, by suitable processes, be entirely converted into certain other kinds of energy; while there are other transformations which can be made only partially complete. These are phenomena which are found to be true by experiment and observation.

Water always tends to flow down a gravity hill to the lowest possible level. *Heat energy at a high temperature elevation always seeks lower temperature levels.* It is the natural tendency for mechanical energy to change into heat energy by means of friction. All of a given quantity of electrical energy may be dissipated in heat through resistance of the conductors. Never yet has a method been found whereby frictional losses, or resistance losses may be completely eliminated. One of the important problems of the engineer is to find means of reducing these losses to the lowest possible value wherever energy transformations are carried on.

With certain forms of energy, transformations to certain other forms may be carried on with a high degree of perfection and completeness. For instance, nearly all of a given quantity of mechanical work may be converted into electrical work. The reverse transformation is easily produced, and at a high efficiency of conversion.

Water, stored at a high gravity elevation, is a much more valuable source of energy than is the same amount of energy stored in a larger quantity of water at a low elevation. Likewise,

a given quantity of heat energy at a high temperature head is more valuable as a source of useful energy than is the same quantity at a low temperature head.

Thus a classification of energy into *high grade* and *low grade* is suggested on the basis of the ease and completeness of conversion into useful forms which is possible. Energy which is capable of complete, or nearly complete conversion; or energy existing at a high potential head is designated as high-grade energy. Energy not readily convertible is called low-grade energy.

Experience shows that there is a universal law by which all high-grade energy naturally tends to degenerate to less useful forms. Thus mechanical work degenerates, by friction, into low-grade heat. High-grade electrical energy, likewise, is dissipated by resistance in the form of low-grade heat. All the ingenuity of scientists and inventors has not as yet devised a method of reversing the process, and by means of friction restore low-grade heat energy to high-grade work energy. Probably it will never happen. It is like expecting, some day, to find a spot on the earth where water will flow up hill without being pushed. Nature simply does not operate that way.

The phenomena described above, may be stated in the form of a corollary to the law of conservation as follows: *In nature, every energy change is accompanied by degradation of energy, in greater or less degree and quantity.*

22. Perpetual Motion Machines of the Second Class. Assume that a mechanical system provided with a fixed store of energy is isolated in an envelope through which energy cannot pass in either direction. Within the mechanism are two devices which are not frictionless. By means of the first, mechanical work is converted into heat. The second device is able to transform all of the heat back into mechanical work. Because the enclosing envelope is a perfect insulator against energy flow, energy in any form cannot escape into space and none can be received. Now the enclosed system, when started functioning, should continue to operate indefinitely. Since no energy is created, the same store of energy being used over and over in the cycle of transformation

from work to heat and heat back to work again, the law of conservation is not violated. However perpetual motion can never be produced by such a system because it is contrary to the principle of degradation of energy.

PROBLEMS

1. An automobile weighing 3800 lbs. is traveling 50 miles per hour on a level highway. Neglecting friction, what uniform retarding force will be required to bring it to a stop in a distance of (a) 10 ft? (b) 50 ft.? (c) 100 ft.?

2. If in Problem 1 the uniform retarding force is in action only for the time required to stop the automobile, what horsepower is generated in each case?

3. A mine shaft is 3500 feet deep. The cage and loaded ore car weigh 7200 lbs. The hoisting cable weighs 2.0 lbs. per ft. The lift is made at a uniform speed of 500 ft. per min. Neglecting friction and acceleration, compute: (a) foot-pounds of work expended for one lift; (b) compute the maximum horsepower required; (c) the average horsepower; (d) the minimum horsepower.

4. Repeat Problem 3 for the case when the weight of the cable varies uniformly from 2.0 lbs. per foot at the end tied to the cage to a weight of 4 lbs. per ft. at the other end.

5. Compute the B.t.u. equivalent of one horsepower hour; of one kilowatt hour.

6. A locomotive pulling a train at a speed of 40 miles per hour develops a drawbar pull of 20 000 lbs. Compute the horsepower developed.

7. It requires 30 horsepower delivered to the rear wheels of an automobile when it is traveling 35 miles per hour. Compute the force acting.

8. If 500 pounds of water per second flowed from a nozzle on a pelton wheel with a velocity of 400 feet per second and it leaves with a velocity of 50 feet per second, what horsepower is delivered by the water?

CHAPTER IV

THERMOMETERS AND TEMPERATURE UNITS

23. Temperature and Temperature Measurements. Temperature is a term used to indicate the state of a body with respect to its hotness or coldness, as indicated by the sensation produced when it is touched with the finger or any other surface of the human body. By means of the sensation produced, one body is said to be hot; another body is said to be cold; or it is said that one body is hotter than another. Hence the idea of a scale of temperature to measure hotness and coldness develops. Quantitative measurement of hotness requires reference to some standard unit of measurement, as is the case for all other kinds of quantitative measurement. When this unit is chosen, the ideas of hotness or coldness may be communicated or recorded on a scientific basis. Henceforth the word temperature will be used as a technical term to convey the primitive meaning of the word hot as well as the idea of a measure of the degree of hotness. When one body is hotter than another, it is said that the first body has a higher temperature. When it is stated that one body has a temperature one hundred degrees higher than a second body, a definite scientific statement comparing the hotness of the two bodies is the result.

Maxwell ¹ defined temperature as follows: "The temperature of a body is its thermal state considered with reference to its power of communicating heat to other bodies."

According to the kinetic theory, temperature is a measure of the velocity and amplitude of the vibration of the molecules and atoms of the substance. If the molecules of a body were absolutely stationary, its temperature would be absolutely zero, and the heat content would be zero. If the molecule were moving slowly through a small amplitude, its temperature would be low. If they move rapidly and through greater distances, the temperature of the body would be high. This brief statement

¹ *Theory of Heat*, p. 32.

probably explains in a crude way why substances expand with an increase in temperature. Since an increase in temperature indicates an increased molecular activity the molecules must have more space in which to dance about.¹

24. Thermometers. The phenomena of expansion of substances when heated was probably first noted and recorded about the year 1600. To Galileo,² the Italian astronomer, is given the credit for the invention and construction of the first thermometer. At first, it was thought that the thermometer measured heat quantities instead of intensity. This notion persisted for more than two hundred years.

The first Galileo thermometer consisted of a hollow glass bulb to which was attached a long slender hollow stem. A slight vacuum was produced in the bulb by heating it gently. Then it was quickly inverted submerging the tip of the open stem in a bath of colored liquid. As the trapped air in the bulb contracted upon cooling, the liquid in the bath rose in the stem a short distance. The stem then remained in this liquid well as in the mercury barometer of to-day. The liquid level in the stem served as an indicator to show the variations in volume of the air trapped in the bulb as the surrounding temperature varied.

This form of thermometer was very sensitive to temperature changes but it was inaccurate due to the effect of changes in atmospheric pressure. Galileo then introduced liquids into the bulb and sealed off the top of the stem. Alcohol seems to have been the liquid which he selected. The first thermometers were graduated by divisions on the stem which represented thousandths of the volume of the bulb. This gave an inconvenient scale when it was desired to compare the indications of different thermometers. Galileo discovered that it is better to select two fixed points on the stem and then divide the interval between them

¹ There are certain substances which expand upon cooling over certain short ranges of their cooling curves. Such substances include water below 39° F. and ice, type metals, and certain alloys of copper and nickel. These are exceptions to the general rule as stated above. The phenomena causing these exceptions are not well understood. But it is believed that chemical or electronic changes may be the cause. At least it is believed that in these exceptional cases there is more than simple thermal change taking place.

² See *Encyclopaedia Britannica*, 11th ed., vol. 13, p. 135.

into an arbitrarily chosen number of degrees. He selected the freezing point of water for the zero mark and he called the temperature of the human body 12 degrees. Thus the first thermometer scale was based on the duodecimal system. In order to secure greater accuracy, Galileo later made this interval 96 degrees.

In 1681, Robert Hooke¹ suggested, and Sir Isaac Newton adopted, the *freezing and boiling points of water* as the most suitable fixed points.

In 1741, Gabriel Fahrenheit, a German mechanician, manufacturer of meteorological instruments, and something of a physicist, improved the thermometer by using mercury as the expanding medium. He also adopted the temperature of a mixture of snow and salt (NaCl) as the zero point. This mixture gave the lowest temperature that could be produced at that time. Fahrenheit believed it to be the lowest attainable temperature. He used the temperature of the human body as the upper fixed point which he called 100 degrees. It was purely accidental that the temperature of the melting point of ice and the boiling point of water turned out to be 32 and 212 degrees, respectively, on this scale. With improvement in methods of manufacture and greater accuracy in graduating thermometers by the Fahrenheit scale it turned out that the normal temperature of the human body is not exactly 100, but is 98.6 degrees. The design of mercury-in-glass thermometer which Fahrenheit developed has persisted practically unchanged to this day. The Fahrenheit scale has proven itself very practical and is used almost universally in all English-speaking countries.

Anders Celsius, Swedish astronomer at the University of Upsala, proposed a thermometer scale which he described in a paper before the Swedish Academy of Science in 1742. On the original Celsius scale the boiling point of water was marked zero and the freezing point 100 degrees. Later, and with the reversed numbering, this scale became the Centigrade scale. The Centigrade scale is employed universally by scientists the world over.

¹ See T. Birch, *History of the Royal Society*, vol. 4.

About 1731, a third thermometer scale was proposed by René Reaumur, a French scientist. Although he published many scientific papers and made some important discoveries regarding the properties of iron and steel, he is now remembered chiefly for the Reaumur thermometer scale. He proposed to use the melting point of water as the zero of the scale. The stem was then divided into degrees each of which represented $1/1000$ of the volume of bulb and stem below the zero mark. It was merely an accident that on this scale and for the expansion rate of the particular quality of alcohol used by him that the boiling point of water turned out to be 80° Reaumur. Mercury thermometers marked 0° R. for the melting point of ice and 80° for the boiling point of water are still much used in Germany and Russia, especially in the beet sugar industry. But they are Reaumur thermometers only in name. The Reaumur scale possesses no especial advantage of any kind to recommend it.

25. Relations of the Temperature Scales. Degrees of temperature as indicated upon the three thermometer scales described above are related to each other as indicated in equations (9) and (10) below.

$$(9) \quad t_f = \frac{180}{100} t_c + 32 = \frac{9}{5} t_c + 32 = \frac{9}{4} t_r + 32,$$

$$(10) \quad t_c = \frac{100}{180} (t_f - 32) = \frac{5}{9} (t_f - 32) = \frac{5}{4} t_r.$$

In these equations

t_c = temperature reading by Centigrade thermometer,

t_f = temperature reading by Fahrenheit thermometer,

t_r = temperature reading by Reaumur thermometers.

Professor John E. Sweet is credited with the following convenient rule for changing Centigrade degrees to corresponding Fahrenheit degrees: Multiply by 2, subtract a tenth of the product, and add 32. In the form of an equation this becomes

$$(11) \quad t_f = 2 t_c - 0.2 t_c + 32.$$

26. Mercury in Glass Thermometers. Mercury has several advantages to recommend it as a thermometer substance. First, it has a fairly high boiling point (680° F.). Second, it does not evaporate appreciably from the top of the thread in the thermometer stem. This causes a serious error in the case of alcohol. By filling the upper part of the thermometer column with an inert gas, such as nitrogen, the evaporation error with mercury can be entirely eliminated. Third, mercury does not wet glass. This avoids errors due to particles adhering to the glass walls of the stem. Fourth, mercury is readily obtained in a high state of purity. Fifth, the very low specific heat and high conductivity of heat of mercury makes it very quick and sensitive in responding to small temperature changes. Because of these advantages it will probably remain in use as the principal means of measuring temperatures ranging from -30° F. to $+800^{\circ}$ F.

However, mercury-in-glass thermometers have certain limitations which must not be exceeded if satisfactory results are to be obtained. At approximately 900° F. thermometer glass begins to soften, causing the thermometer to lose its calibration. The volume of the mercury well is not constant. After each measurement of a temperature of 200° F. or above, the mercury well is permanently enlarged by a slight amount. This makes all subsequent indications of the thermometer slightly too low. This error is called depression of the ice point. This fault may be greatly reduced but not eliminated by aging the thermometer after it is made up, but before the scale is placed upon it. Formerly, the best thermometers were aged as long as ten or fifteen years. But now, methods of aging by heat treatment in annealing ovens for a few days accomplishes more aging effect than storage for years formerly did. Thermometer indications are due to the difference in the rate of expansion of mercury and glass. Neither material has an expansion rate that is exactly constant at all temperatures. Hence for refined and precise measurements much tedious calibration work must be done determining the magnitude of the errors for this type of deviation.

Nearly all chemical thermometers are graduated to indicate correct temperatures when the entire thermometer is submerged in a bath at the temperature under consideration. Engineers nearly always use a thermometer with only the mercury bulb and a short section of the stem submerged. In that case the emergent part of the stem is exposed to the temperature of the surrounding atmosphere. In measuring steam temperatures which range between 200° F. and 800°

F. the exposed stem and mercury column are cooler than the steam; hence the thermometer indications are too low. This correction may amount to as much as 20° F. in extreme cases. An approximate correction for the error caused by the exposure of the stem may be computed as follows.¹ An auxiliary thermometer is placed in contact with the main thermometer so adjusted that the well of the auxiliary thermometer is located at about the middle of the exposed mercury column of the main thermometer. The auxiliary thermometer will indicate the approximate mean temperature of the exposed mercury column. Then

$$(12) \quad K = c(t - t_e)(t - t_a).$$

In equation (12)

c = a coefficient. For the Fahrenheit scale $c = 0.000085$; for the Centigrade scale $c = 0.00015$.

K = correction in degrees,

t = main thermometer reading in degrees,

t_a = auxiliary thermometer reading in degrees,

t_c = correct temperature,

t_e = reading on main thermometer in degrees at which stem emerges from the bath.

Then

$$(13) \quad t_c = t + K.$$

In equation (12) the coefficient c represents the difference in expansion between mercury and glass for one degree of temperature difference and for one degree of exposed stem. Its value varies with the kind of glass used in making the thermometer. For Fahrenheit thermometers the value varies between 0.00006 and 0.0001 with 0.000085 as an average value; 0.00015 is an average value for Centigrade thermometers. Stem corrections are unnecessary if the thermometer is scaled for partial submersion when it is made.

Thermometers made from fused quartz but having the form of mercury-in-glass thermometers are now available. They are filled above the mercury with some inert gas (usually nitrogen or CO₂) at a pressure of about 60 atmospheres. This type of thermometer may be used for measuring temperatures up to about 1400° F.

27. Pyrometers. For very precise and accurate temperature measurement at ordinary temperatures, and for high temperatures,

¹ See *United States Bureau of Standards Circular No. 8*.

electrical pyrometers are usually employed. They operate on two different principles. One type depends upon the fact that the electrical resistance of metals varies in a more or less regular and definite way with changing temperature. Some metals are much superior to others in this respect. Platinum wire has a fairly large resistance variation per degree of temperature change and the variation is very uniform over a wide range, hence it is one of the best metals to use in the resistance pyrometer.

The other type of pyrometer takes advantage of the phenomenon that when the junction of two dissimilar metals is heated a measurable voltage, which is proportional to the temperature, is generated. A junction of platinum and an alloy of platinum and rhodium has been found to be about the best for high temperature work up to 3000° F. A copper-constantan junction is used most successfully for low temperature work below atmospheric temperature. Constantan is an alloy composed of sixty per cent copper and forty per cent nickel. The voltage generated by a thermocouple may be measured by a galvanometer or by a delicate millivoltmeter.

28. Fixed Temperatures. It has been found that the melting points and boiling points of certain substances are very constant under certain conditions which may readily be reproduced. The substance should be readily obtained in a pure state. It should not readily oxidize, and it should be easily obtainable in reasonably large quantities for making thermometric calibrations. The melting point of ice and the boiling point of water are the two fixed points most frequently employed for ordinary temperature ranges. In using any substance as a means of determining a fixed temperature, there are always certain precautions to be observed. For example, H_2O ice melts at 32° F., but ice made from pure distilled water should be used. Great care should be taken to see that it is placed in a chemically clean receptacle and to be sure that the ice does not at any time become contaminated with salts of any kind. The boiling point of water is affected by the kind of container, by the pressure of the atmosphere, and many other conditions. But it has been found that the temperature of water vapor collected over boiling water is practically unaffected by anything except the pressure of the vapor. If the vapor pressure is exactly 29.921 inches of mercury,

the temperature of the vapor is exactly 212° F. The properties and peculiarities of any substance used must be studied, and any condition causing variations in reproducing any temperature under consideration must be carefully controlled.

A list of substances which have been found suitable for use in thermometer calibrations is given in Table 7, arranged in the order of ascending temperatures.

TABLE 7
FIXED TEMPERATURES USED IN CALIBRATING THERMOMETERS

No.	SUBSTANCE	FIXED POINT	TEMPERATURE DEG. F.
1.	Oxygen	Boiling pt.	-297
2.	Mercury	Melting pt.	- 37.8
3.	Water	Melting pt.	32.0
4.	Water	Boiling pt.	212.0
5.	Tin	Melting pt.	449.4
6.	Sulphur	Boiling pt.	832.2
7.	Aluminum	Melting pt.	1217.6
8.	Silver	Melting pt.	1760.4
9.	Nickel	Melting pt.	2646.
10.	Platinum	Melting pt.	3191.

29. Law of Thermometry. All temperature measurements are based upon the following principle which was first stated by Maxwell.¹ *If, of three bodies (A, B, and C), A is at the same temperature as B and also at the same temperature as C, then B and C must be at the same temperature.* That is, all three bodies are in thermal equilibrium. This is another of Nature's laws which cannot be proved *a priori* by mathematical reasoning. It can only be proved by experimental methods carried out under every conceivable condition. Thus a thermometer when inserted into a vapor, collected over boiling water at a pressure of one standard atmosphere, may indicate a temperature of 212° F. If, upon inserting the same thermometer into a barrel of hot syrup it again indicates 212° F., the vapor and syrup are known to be at the same temperature or in thermal equilibrium; therefore their molecular activities are known to be equal. If the two substances

¹ *Theory of Heat*, p. 32.

were mixed without radiation losses taking place, and then the thermometer is placed in the mixture, it would again register 212° F.

PROBLEMS

1. (a) Convert the temperature of the boiling point of oxygen, as given in Table 7, into the corresponding Centigrade temperature. (b) Repeat for water, sulphur, and platinum.

2. Determine the temperature at which the Fahrenheit and Centigrade thermometer scales indicate the same reading.

3. Determine the temperature at which the Fahrenheit and Reaumur thermometer scales indicate the same reading.

4. Convert the following centigrade temperatures into the corresponding Fahrenheit temperatures: 10° ; 20° ; 37° ; 65° ; and 400° .

5. Convert the following Fahrenheit temperatures into the corresponding Centigrade temperatures: 0° ; 60° ; 150° ; 350° ; and 500° .

CHAPTER V

FUNDAMENTAL LAWS OF GASES

30. Cardinal Properties. A cardinal property of a substance is one which can be determined by an immediate examination of the substance. It is any property which is defined by the immediate state of the substance. Properties which do not fix the present state of a substance, or whose values are not determined by the present state, are not cardinal properties of that substance.

Cardinal Properties of Gases and Their Symbols	{	Pressure..... P
		Volume..... V
		Temperature..... T
		Entropy..... N
		Internal Energy... E_i

The cardinal properties of *perfect gases* are so related, mathematically, that if the value of any two properties for a given condition are known all the others may be readily computed. This fact explains in a measure why these properties are so important, and why they are called cardinal. They are, in a way, the concrete blocks with which the structure of thermodynamics is constructed. The supporting evidence for these statements will be developed in the following pages. Again, this is one of those facts which cannot be wholly proven by mathematical reasoning.

31. Boyle's and Mariotte's Law. From 1657 to 1660, Boyle worked with Robert Hooke, discoverer of Hooke's law, studying the air pump. They made some important improvements in the pump, and in 1660 Boyle published a paper on their work, *New Experiments, Physico-Mechanical, Touching the Spring of Air and Its Effects*. In a printed reply to some criticisms made on the air pump proper by a Jesuit in France, Francisus Linus, Boyle enunciated the general principle since known as Boyle's law.

In Paris, about this time Edme Mariotte was investigating the properties of gases also. He knew nothing of Boyle's work or his publications. Mariotte, however, was not as prompt in publishing his results as Boyle. It was in 1676, or sixteen years after Boyle's paper appeared, before Mariotte published his paper in which he again stated this great principle, discovered independently by him.

This principle is now generally given the double designation, Boyle's or Mariotte's law. The proof of this law was originally based entirely upon experiment, but later it was developed mathematically from the kinetic theory of gases.¹

Boyle's and Mariotte's law may be stated briefly as follows: *At constant temperature, the volumes of a constant mass of any gas are inversely proportional to the pressures.* In the form of an equation the law becomes:

$$(14) \quad V \sim \frac{1}{P} \text{ (temperature constant),}$$

or

$$(15) \quad V = \frac{C}{P},$$

and

$$(16) \quad PV = P_1V_1 = P_2V_2 = C.$$

Also the following may be stated:

$$(17) \quad V = Mv.$$

The symbols used above represent quantities as follows:

$\sim$ should be read, "varies as,"

C = a constant,

P = the gas pressure measured in any system of units,

V = the gas volume measured in the same system of units as P ,

v = volume of 1 pound of gas in cu. ft., or specific volume,

M = the weight or mass of gas under consideration.

The symbols used without subscripts, refer to the quantities in general. The subscripts ₁, ₂, etc., refer to values under particular conditions as may be specified.

¹ See Encyclopaedia Britannica, 11th edition, vol. 18, p. 657.

The relations expressed by equations (14) to (17) are true for any system of units so long as that system is used consistently. In practically all the succeeding pages of this book, the English system (foot, pound, second) will be understood.

The pressure of one standard atmosphere is 29.921 inches of mercury when the mercury column is at the temperature 32° F. This is equivalent to 14.969 pounds per square inch or $14.696 \times 144 = 2116.2$ pounds per square foot.

By careful laboratory measurements, the volume of one pound of air when determined at a temperature of 32° F. and a pressure of one standard atmosphere is 12.39 cubic feet. When these values are substituted in equations (16) and (17) the result is

$$(18) \quad P_a v_a = 2116.2 \times 12.39 = 26,220,$$

in which the subscript a , refers to standard atmosphere conditions.

32. Law of Charles and Gay-Lussac. Jacques A. Charles, 1746–1823, French mathematician and physicist, discovered the law of the expansion of gases with changing temperature in 1787, but he did not consider it of sufficient importance to publish it. It should be noted that this discovery was made more than 100 years after Boyle announced his law.

Joseph L. Gay-Lussac, 1778–1850, French chemist and physicist, was the elder son of Antoine Gay, procurer du roi, at Port-de-Noblac. The father took the name, Lussac, from a small property which he owned near St. Leonard. Joseph became a laboratory assistant to C. L. Berthollet. His early investigations were carried out upon the physical properties of gases. In 1802, Joseph Gay-Lussac published his first scientific paper in which he announced his independent discovery that "*At constant pressure, gases expand at a uniform and constant rate between 0° C. and 100° C.*" He explained in the paper that 15 years earlier Charles had noted the same phenomenon. He explained further that he had learned of Charles' result only by accident after he had made the discovery himself. Gay-Lussac and John Dalton, independently, measured the rate of change of volume with change of temperature when the pressure is held constant. They found that the volume increases about 1/485 of the volume at 32° F. for each degree

Fahrenheit rise. They also found that, when the volume is held constant, the pressure changes at the same rate.

By more extended and refined research, Henri V. Regnault, between 1840 and 1858, found that the *rate of change is about 1/491.6 of the pressure or volume at 32° F.* Regnault was the first to note that the rate of change was not exactly constant throughout the possible temperature range and that the expansion rate is slightly different for different gases.

33. Absolute Zero of Temperature. If the rate of change remained constant for all temperatures and for all gases, the volume of a gas measured at 32° F. would apparently become zero upon lowering the temperature 491.6 degrees, provided the pressure remained constant. Or if the volume were held constant, the pressure would apparently become zero at 491.6° degrees, below 32° F. These conditions are illustrated in Figures 1 and 2.

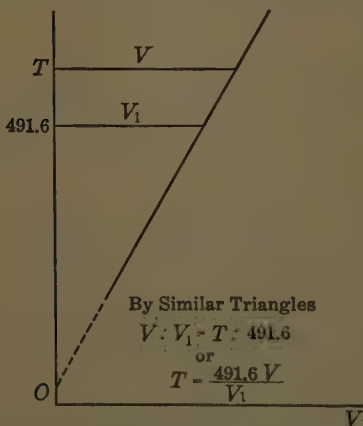


FIG. 1.—TEMPERATURE—VOLUME CHANGE (PRESSURE CONSTANT)

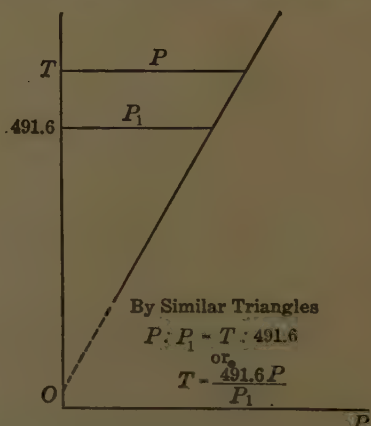


FIG. 2.—TEMPERATURE—PRESSURE CHANGE (VOLUME CONSTANT)

Experience in the laboratory has taught that it is impossible to reduce the volume of a gas to zero. At some point down the temperature scale every gas changes to the liquid state. In that state the volume cannot well be reduced below a certain minimum.

By the kinetic theory, the molecules of gases are traveling through space in straight lines at high velocities relatively so far

removed from each other that there is practically no mutual attraction between them. They only change their directions when they collide with each other or with the walls of the container. After each collision they rebound with undiminished velocity and energy if the gas were a perfect one. When the volume of a given mass of gas is reduced by increasing the pressure or by a reduction of temperature, the molecules are forced closer together. This reduction in volume must apparently stop when the molecules are forced into close contact with each other. That state of affairs is distantly approximated when a substance is reduced in volume from the gaseous to the liquid state. However, in the liquid state, there still remains considerable free space between the molecules. But obviously, a condition would eventually be reached when the temperature becomes sufficiently reduced to force the molecules into close contact with each other. Then any further lowering of the temperature would have no more effect on the volume occupied.

The phenomenon of pressure is explained, by the kinetic theory, as being produced by the summation of all the effects of the incessant bombardment of the walls of a gas container by vast numbers of molecules in their confused migratory motions. As the temperature falls, the volume being held constant, the movements of the molecules become less rapid until finally at a temperature 491.6° F. below 32° the pressure drops to zero with the molecules presumably at rest. Since temperature is a manifestation due to molecular and atomic activity, when these movements cease, the temperature must reach a value below which it cannot go. That point is known as the Absolute Zero Temperature. Temperatures measured above that point are called absolute temperatures.

Since heat energy is due to the kinetic energy of molecules and atoms in motion, it follows that heat energy must be zero when these motions cease at absolute zero temperature. This is one of the fundamental notions upon which the science of thermodynamics is built.¹

¹ Recent investigations into the fundamental structure of matter, radio activity, and kindred subjects indicate that many of these phenomena slow down at low temperatures and may even cease altogether at absolute zero temperature. For

Absolute zero temperature has never yet been produced in the laboratory. But in 1913, Kamerlingh Onnes, director of the Physical Laboratory at Leyden, succeeded in liquifying helium. To do this, a temperature only 8.1° F. above absolute zero was reached.

The relation between ordinary thermometer scale indications and temperature indications on the absolute scale is expressed by the following equations:

$$(19) \quad T_f = t_f + 459.6,$$

$$(20) \quad T_c = t_c + 273.1,$$

in which

T_f = absolute temperature in degrees Fahrenheit,

t_f = thermometer reading in degrees Fahrenheit,

T_c = absolute temperature in degrees Centigrade,

t_c = thermometer reading in degrees Centigrade.

In general, throughout the remainder of this work, the symbol T will be used to represent absolute temperatures and t thermometer indications.

34. Equation of State. Charles's law may now be stated in the form of equations which are more convenient for purposes of computation. For constant pressure, equation (21) holds.

$$(21) \quad V = \frac{TV_1}{491.6} = \frac{TV_1}{T_1}.$$

When the volume is held constant, equation (22) gives the relation

$$(22) \quad P = \frac{TP_1}{491.6} = \frac{TP_1}{T_1}.$$

In equations (21) and (22)

P_1 = the initial pressure of the gas in pounds per square foot
when the temperature is 32° F.,

P = the final pressure in pounds per square foot at the
final temperature T° F.,

this reason there is justification for the belief that heat energy is a manifestation due not only to the movements of molecules but also to some extent of interatomic activities.

V_1 = the initial volume of the gas in cubic feet when the temperature is 32° F. ,

V = the final volume of the gas at the final temperature $T^\circ \text{ F.}$,

T_1 = the initial temperature of the gas in absolute degrees Fahrenheit. It may be $461.6^\circ \text{ F. abs.}$ or any other temperature,

T = the final temperature of the gas in absolute degrees Fahrenheit.

Equations (16), (21), and (22) may be written as follows:

$$(23) \quad PV = P_1 V_1,$$

$$(24) \quad \frac{V}{T} = \frac{V_1}{T_1},$$

and

$$(25) \quad \frac{P}{T} = \frac{P_1}{T_1}.$$

When equations (23) and (24) and (25) are multiplied together, the result is

$$(26) \quad \frac{P^2 V^2}{T^2} = \frac{P_1^2 V_1^2}{T_1^2},$$

from which

$$(27) \quad \frac{PV}{T} = \frac{P_1 V_1}{T_1}.$$

If for some temperature, as 32° F. equal to $491.6^\circ \text{ F. absolute}$, the values of P_1 and V_1 may be measured, the relation $P_1 V_1 / T_1$ acquires a definitely fixed value which is always constant. Then equation (27) becomes

$$(28) \quad \frac{PV}{T} = C \text{ (where } C \text{ is a constant).}$$

Substituting equation (17) in (28) one obtains the result

$$(29) \quad \frac{PMv}{T} = MR = C.$$

Therefore

$$(30) \quad \frac{PV}{T} = MR,$$

and

$$(31) \quad PV = MRT.$$

Or, for one pound of the substance,

$$(32) \quad Pv = RT.$$

In equations (28) to (32) the interpretation of the symbols used is as follows:

- C = a constant for M lbs. of gas,
- M = the lbs. of gas whose volume is V cu. ft.,
- R = a constant for 1 pound of gas,
- v = the volume of one pound of gas in cu. ft., or specific volume.

The other symbols represent quantities as explained on pages 42 and 43.

Equation (32) is known as the *equation of state* for a gas. It is perfectly general for all conditions of temperature, pressure, and volume for a perfect gas so long as it is applied to one pound of gas. For weights other than one pound, equation (31) applies. The quantity, R , is a constant which depends for its value upon the nature of the gas and the system of units employed in measuring P , V , and T .

35. Value of the Constant R . By careful and repeated laboratory measurements for air, it has been found that at a pressure of one standard atmosphere (29.921 inches of mercury or 14.696 pounds per square inch) and at a temperature of 32° F., one cubic foot of air weighs 0.080712 pounds. Therefore, one pound of air at standard conditions of temperature and pressure has a volume of 12.39 cubic feet. By substituting these values in equation (32), the value of R for air is obtained.

$$(33) \quad R = \frac{P_1 V_1}{T_1} = \frac{14.696 \times 144}{0.080712 \times 491.6} = 53.34.$$

Hence the value of R , for air in English units, is 53.34. Values for other gases are given in Table 8, page 67.

36. Perfect Gas. Continued and extended research has shown that no gas which complies exactly with Boyle's and Charles'

laws exists in Nature. Under extreme conditions of pressure and temperature, all gases show more or less pronounced deviations and at certain fixed temperatures and pressures all gases are reduced to the liquid state. Conceivably, a hypothetical gas could exist which would conform exactly with the above-mentioned laws, thus fulfilling the conditions of the equation of state (equation (32)). Such a hypothetical gas is called a perfect gas.

37. Avogadro and His Law. Amedeo Avogadro, 1776–1856, was an Italian professor of higher physics at Turin University. He published many papers on the subject of heat and its manifestations. In 1811, he published the law which will perpetuate his name as long as the scientific era endures. Avogadro's law is, *At the same temperature and pressure all gases contain the same number of molecules per unit of volume.* Hence the weights of gases, per unit of volume, must be proportional to their molecular weights.

Avogadro's law furnishes a basis upon which to compute the value of the constant R , for any gas, provided the molecular weight of the gas is known. The molecular weight and specific volume of some standard gas must also be known. Oxygen has probably been studied more exhaustively, and the values of its properties are more accurately established than for any other gas. Hence oxygen is generally selected as a standard for comparison.

38. Molecular Gas Constant. The weight of a gas per unit of volume is its density. Hence the following equation may be said to express Avogadro's law:

$$(34) \quad m : m_1 : m_2 = \frac{1}{v} : \frac{1}{v_1} : \frac{1}{v_2},$$

from which

$$(35) \quad mv = m_1v_1 = m_2v_2,$$

m = molecular weight of any gas,

v = the volume of 1 pound of the same gas.

The subscripts $_1$ and $_2$ refer to any specified gases. The result of laboratory research indicates that the molecular weight of oxygen is exactly 32 when the molecular weight of hydrogen is taken at

2.016. At 32° F. and a pressure of one standard atmosphere, the volume of one pound of oxygen is 11.212 cubic feet. Putting these values in equation (35) the result is

$$(36) \quad mv = 32.00 \times 11.212 = 358.78,$$

and

$$(37) \quad v = \frac{358.78}{m}.$$

Substituting (37) in the equation of state, (32), one obtains

$$(38) \quad R = \frac{Pv}{T} = \frac{2116.2}{491.6} \times \frac{358.78}{m} = \frac{1545}{m},$$

and

$$(39) \quad mR = m_1R_1 = m_2R_2 = 1545.$$

The quantity, 358.78, of equation (36) is known as the *molecular gas volume*. Likewise, the constant in equation (39) is called the *molecular gas constant*. The volume of one pound of any gas at standard atmospheric pressure and 32° F. may always be found by equation (37). And the value of R for any gas is likewise found from equation (38) provided the molecular weight is known.

39. Dalton's Law of Partial Pressures. John Dalton (1766–1844), English chemist, first conceived the idea, about 1800, that it should be possible to reduce all gases to the liquid state. He was also one of the first to believe in the atomic structure of matter. In 1803, he published a paper, *Absorption of Gases by Water and Other Liquids*, which probably contained his most important observation. It has since been known as Dalton's law. Briefly stated it is: *Every gas appears to be a vacuum for every other gas*. A more complete statement is as follows: The total pressure exerted upon the walls of a container, in which there is a mixture of gases, is equal to the sum of the pressures produced by each constituent gas if each one occupied the container alone at the temperature of the mixture. The gases must not combine chemically when mixed. Dalton's law may be stated in the form of an equation as follows:

$$(40) \quad (p_1 + p_2 + p_3 + \dots) = P.$$

Dalton's law may also be embodied in the equation of state as follows:

$$(41) \quad (p_1 + p_2 + p_3) V = (m_1 r_1 + m_2 r_2 + m_3 r_3) T.$$

The value of R_m for a gas mixture may now be obtained from equations (31), (40), and (41) as follows:

$$(42) \quad M R_m = m_1 r_1 + m_2 r_2 + m_3 r_3 + \dots,$$

$$(43) \quad R_m = \frac{m_1 r_1}{M} + \frac{m_2 r_2}{M} + \frac{m_3 r_3}{M} + \dots$$

In equations (40) to (43) inclusive the symbols used have the following significance:

P = total pressure in lbs. per sq. ft. absolute of the gas mixture in the container,

V = total volume of the container in cu. ft.,

M = weight of the mixture in pounds,

T = absolute temperature of the mixture in degrees Fahrenheit,

p_1, p_2, p_3 , etc. = partial pressure in lbs. per sq. ft. absolute each gas would produce if it occupied the container alone.

m_1, m_2, m_3 , etc. = weights in lbs. of each gas in the mixture,

r_1, r_2, r_3 , etc. = the respective gas constants for each gas.

PROBLEMS

1. Compute the value of R for hydrogen, nitrogen, and carbon dioxide. Use the molecular gas constant. Take molecular weights from Table 8, page 67.

2. Compute the value of the molecular weight of air. (Consider air to be a mixture made up of nitrogen 76.8 per cent and oxygen 23.2 per cent by weight.)

3. Compute the value of R for air in terms of kilogram-meter-centigrade units.

4. Develop an equation to use for the conversion of absolute Fahrenheit degrees to absolute Centigrade degrees.

5. A quantity of air at 250° F. and a pressure of 150 lbs. per sq. in. has a volume of 12 cu ft. Find the temperature when the volume has increased to 150 cu. ft. at a pressure of 14.7 lbs. per sq. in. Find the weight of the air. (Assume that the air follows the laws of a perfect gas.)

6. Compute the value of the molecular gas constant in kilogram-meter-centigrade units.

7. Repeat Problem 6 for molecular gas volume.

8. A closed tank, whose fixed volume is 500 cubic feet, contains air at a pressure of 145 lbs. per sq. in. abs. and at a temperature of 100° F. Air is then removed from the tank until the pressure falls to 35 lbs. per sq. in. abs. and the temperature drops to 60° F. How many pounds of air were removed? How many pounds remained in the tank?

9. Suppose the air pressure in the tank of Problem 8 was initially 50 lbs. per sq. in. abs. and the initial temperature was 75° F. Find the new pressure when the temperature falls to 32° F., the weight of air in the tank remaining constant.

10. The pressure in an automobile tire is 40 lbs. per sq. in. abs. at 60° F. What is the pressure when the temperature is 130° F. (The volume of the tire and the weight of air in the tire are assumed to remain constant.)

11. At 32° F. and a pressure of 29.921 ins. of mercury, 1 cu. ft. of oxygen weighs 0.0892 lb. Its molecular weight is 32.00. Compute the volume occupied by 5 lbs. of CO₂ at 100° F. and a pressure of 30.50 ins. of mercury. Repeat for 10 lbs. of hydrogen at 60° F. and 50 lbs. per sq. in. abs. (One cubic inch of mercury at 32° F. weighs 0.491 lbs.)

12. From the molecular formulas of the hydrocarbons listed below, compute their specific volumes, their weights per cu. ft., and the values of R at 32° F. and atmospheric pressure: ethylene C₂H₄, ethane C₂H₆, propylene C₃H₆, propane C₃H₈.

CHAPTER VI

HEAT MEASUREMENT—SPECIFIC HEAT

40. Standard Heat Unit. When heat energy is imparted to a substance, the velocity and amplitude of the movements of the molecules are both increased. This change in molecular activity is made apparent to an observer by an increase in the temperature as indicated by a thermometer. If the same quantity of heat energy be imparted to different substances the temperature increases indicated may be different. This is due to the fact that the quantity of energy taken up by the different substances varies as the mass of the molecules and it also varies as the difference of the squares of the respective velocities after and before the addition of heat energy. Expressed in the form of an equation it becomes:

$$(44) \quad Q \sim M(S_2^2 - S_1^2),$$

where

M = mass of a molecule,

Q = quantity of heat energy

S_1 = mean velocity of molecules before addition of heat energy,

S_2 = mean velocity of molecules after addition of heat energy.

Hence to measure heat quantities as well as temperatures scientifically there must be a standard heat unit to correspond to the standard unit of temperature.

A standard mass, a standard molecule, and a standard change in molecular activity must be employed in defining a standard heat unit. Scientists and engineers have arbitrarily chosen and agreed upon these standards. The scientists have adopted the mass of one gram's weight. The standard molecule agreed upon by both scientists and engineers is that of pure water (H_2O). The standard change of molecular activity agreed upon by the

majority of the scientists is that indicated by a temperature increase from 20 to 21 degrees measured on the Centigrade thermometer scale. Some scientists prefer to use other temperature limits. The range from 0° C. to 1° C. has been used by some. Others have used the range from 14.5° C. to 15.5° C. But the limits 20° C. and 21° C. are probably more generally used than any other. The quantity of heat energy required to raise the temperature of 1 gram of pure water from 20° C. to 21° C. has been named the *calorie* or the *gram-calorie for 20° C.* The energy required to heat 1000 grams of water from 20° C. to 21° C. is called a great calorie or *kilogram-calorie*.

41. British Thermal Unit. Engineers, in English-speaking countries, use the unit of heat based upon 1 pound, 1 degree Fahrenheit, and the water molecule. The change in temperature much used formerly, was that near the greatest density of water *i.e.* 39° F. to 40° F. This unit is called the *British thermal unit*. The standard abbreviation for it is *B.t.u.*

From time to time, other temperature limits such as 32° F. to 33° F. and 59° F. to 60° F. have been proposed by different authorities, but since about 1907, when new steam tables were first published by Marks and Davis, it has become general practice among engineers to adopt the use of a *mean B.t.u.* This is 1/180 of the heat energy required to change the temperature of one pound of distilled water from 32° F. to 212° F. In a similar manner a *mean calorie* has been defined, and much used by scientists. The modern tendency is to use the mean calorie and the mean B.t.u. as the standard heat units.

From the relations between the gram, pound, degree Centigrade, and degree Fahrenheit it follows that

(45)

$$1 \text{ B.t.u.} = 252 \text{ calories.}$$

42. Specific Heat. If careful observations are made, it will be found that just twice as much heat energy is required to change the temperature of two pounds of water one degree, say from 32° F. to 33° F., as is required by one pound. That is, the heat energy required to raise the temperature of a given substance a

given number of degrees varies directly as the mass of the substance heated. But to change the temperature of one pound of mercury one degree, say from 32° F. to 33° F., requires only about one-thirtieth of the heat energy absorbed by one pound of water for the same temperature range. The ratio of the two quantities of heat required to raise two different substances one degree of temperature is called *specific heat*. This may be more concisely stated in the form of an equation as follows:

$$(46) \quad c = \frac{\text{heat to raise 1 lb. of mercury from 32° F. to 33° F.}}{\text{heat to raise 1 lb. of water from 32° F. to 33° F.}}$$

The ratio represented by the symbol c is called the specific heat of mercury. Or more exactly, it requires 0.033 B.t.u. to change the temperature of one pound of mercury from 39° F. to 40° F. It also requires one B.t.u. to change the temperature of one pound of water from 39° F. to 40° F. Then the value of the specific heat of mercury at 39.5° F. is

$$(47) \quad c = \frac{0.033}{1.000} = 0.033$$

The values of the specific heats of all liquids and solids are based upon the heat capacity of water at some standard temperature taken as unity. Specific heat values of gases are based upon the heat capacity of air being considered equal to unity. The specific heat value of practically every solid and every liquid is less than 1.00.¹ The value of the specific heat of any substance is the same whether the heat unit employed is the calorie or the B.t.u.

43. Heat and Temperature. The words *heat* and *temperature* are often improperly used synonymously. This is a gross error which should never be committed by engineers or scientists. The term heat or heat energy refers to a quantity of energy. It may be considered analogous to a quantity of water in a city main. The term temperature may then be considered analogous to the pressure of the water in the pipe line. Analytically the

¹ See tables of specific heat values for different substances in Marks' *Mechanical Engineers' Handbook*, or Landolt-Börnstein's *Physikalisch-Chemische Tabellen*.

relation between heat quantities and temperature values may be represented thus:

$$(48) \quad dQ = cM dT = cM dt.$$

Then

$$(49) \quad c = \frac{dQ}{M dT} = \frac{dQ}{M dt},$$

and

$$(50) \quad Q = \int_{T_1}^{T_2} cM dT = \int_{t_1}^{t_2} cM dt.$$

If c and M are constants, equation (50) becomes

$$(51) \quad Q = cM \int_{T_1}^{T_2} dT = cM(T_2 - T_1) = cM(t_2 - t_1),$$

where

c = specific heat,

M = lbs. of the substance heated,

Q = a quantity of heat in B.t.u. or calories,

T_1 = initial absolute temperature,

t_1 = initial temperature, thermometer reading,

T_2 = final absolute temperature,

t_2 = final temperature as indicated by thermometer scale reading.

For most substances, it has been found that the value of c is not a constant but it increases as the values of T or t increase. This increase may be represented by equations as follows:

$$(52) \quad c = a + bt,$$

or

$$(53) \quad c = A + BT.$$

In equations (52) and (53) a , A , b , and B are constants. These equations represent straight lines which may hold for small ranges of temperature. For greater accuracy or over wide temperature ranges, the variation of specific heat does not follow a straight line law, but follows a curve which requires higher powers of t or T to correctly represent it, as in equations (54) and (55) below.

$$(54) \quad c = a \pm bt \pm \frac{e}{t^2} \pm ft^3 + \dots,$$

and

$$(55) \quad c = A \pm BT \pm \frac{E}{2} T^2 \pm FT^3 + \dots.^1$$

In equations (48) to (55) inclusive, c is called the *instantaneous specific heat* at the temperature t or T . The coefficients of t and T are constants. Substituting (54) in (48), one obtains

$$(56) \quad dQ = M(a \pm bt \pm et^2 \pm ft^3)dt.$$

Integrating (56) between the limits 0 and t degrees, one obtains

$$(57) \quad Q = M \int_0^t (adt \pm btdt \pm et^2dt \pm ft^3dt),$$

$$(58) \quad Q = M \left(at \pm \frac{b}{2} t^2 \pm \frac{e}{3} t^3 \pm \frac{f}{4} t^4 \right).$$

Equations (57) and (58) are special cases because the temperature limit 0 is a unique value. It can be readily seen that some *mean value* of c such as $\bar{c}$ multiplied by the temperature range from 0 to t will give Q . Hence

$$(59) \quad Q = \bar{c}M(t - 0) = M \left(at \pm \frac{b}{2} t^2 \pm \frac{e}{3} t^3 \pm \frac{f}{4} t^4 \right),$$

or

$$(60) \quad Q = \bar{c}M(T - 0) = M \left(AT \pm \frac{B}{2} T^2 \pm \frac{E}{3} T^3 \pm \frac{F}{4} T^4 \right).$$

The symbol, $\bar{c}$, represents the *mean specific heat* between 0 and t degrees of temperature. Equations (59) and (60) are true only for the special condition that the initial temperature is zero. However it is possible to derive an equation similar in form which will be perfectly general.

44. Law of Mixtures. An equation of heat quantities may be formulated to express the situation as regards heat exchanges when a number of substances at different temperatures are mixed, or which may readily be brought into thermal equilibrium by mutual contact. When no extra heat energy is added, and no heat energy is used to perform work, or is lost by radiation, the equation of heat is

$$(61) \quad \bar{c}_1 M_1 (t_1 - t_f) + \bar{c}_2 M_2 (t_2 - t_f) + \bar{c}_3 M_3 (t_3 - t_f) + \dots = 0.$$

¹ For recently determined values of the constants, see Table 10, p. 72 herein.

The symbols in (61) represent the following items:

- $\bar{c}_1, \bar{c}_2, \bar{c}_3$, etc. = the respective mean specific heats of a number of substances over the specified temperature ranges,
 M_1, M_2, M_3 , etc. = the respective weights of the substances,
 t_1, t_2, t_3 , etc. = the respective initial temperatures of the substances before mixing,
 t_f = the final temperature of the mixture as shown by the thermometer reading.

Equation (61) is frequently used to determine the unknown mean specific heat of a substance over a short range of temperature by mixing predetermined quantities of the substance whose mean specific heat is unknown with one whose specific heat is known. In that procedure all the factors in equation (61) may be measured except the unknown specific heat which may be evaluated by solving the equation.

PROBLEMS

1. Show how the relation expressed in equation (45), p. 50 is obtained.
2. Develop an equation similar to (59), p. 53 which will be perfectly general.
3. Compute the B.t.u. required to raise the temperature of 9 pounds of carbon dioxide from 32° F. to 3000° F. if $c = 0.210 + 7.4 \times 10^{-5}t - 1.8 \times 10^{-8}t^2$. What is the value of the mean specific heat, $\bar{c}$, for this temperature range?
4. When 0.8 gram of gasoline is burned in a calorimeter, the temperature of 2600 grams of water rises from 68.00° F. to 74.55° F. Compute the B.t.u. which 1 lb. of gasoline generates.
5. The following substances were mixed in the proportions and at the initial temperatures indicated below. The final temperature of the mixture was 131.5° F. Compute the mean specific heat of the alcohol.

Substance	M	$t^\circ\text{F.}$	$\bar{c}$
Mercury	7.0	150	0.033
Water	10.0	180	1.00
Alcohol	15.0	75	—

6. If an automobile engine transforms 15 per cent of the heat developed in the cylinder into useful work at the rear wheels, how many B.t.u. are required in the engine cylinder to furnish 1 horsepower hour?

7. Using the results of Problems 4 and 6, compute the number of pounds of gasoline required to develop 1 horsepower hour at the wheels.

8. Compute the increase in volume when one cubic foot of each of the following substances is heated from 32° F. to 500° F. ; cast iron, water in the liquid state, and air. Calling the increase in volume of cast iron 1.00, what is the increase, in terms of the increase for cast iron, for each of the other substances? (Take the necessary coefficients from any engineers' handbook.)

9. The nitrogen of the air is inert and useless in producing combustion in a boiler furnace. Compute the B.t.u. carried away in the stack gases by one pound of nitrogen which has been heated from 80° F. to 500° F. (First determine the value of the mean specific heat, c_p , from Table 10, p. 72.)

10. If one pound of carbon whose heating value is 14 600 B.t.u. is burned in a boiler furnace at constant pressure producing 8.83 pounds of nitrogen and $3\frac{2}{3}$ pounds of CO_2 , compute the theoretical maximum temperature produced. (Assume the equations for c_p for N and CO_2 of Table 10, p. 72 are correct for this problem; also assume that the room temperature is 70° F.)

CHAPTER VII

TRANSFORMATION OF HEAT ENERGY INTO MECHANICAL WORK AND SPECIFIC HEATS OF GASES

45. Mechanical Work from Heat Energy. When any substance is heated at constant pressure its volume will change (usually increase). The volume increase will take place against enormous resistance if only sufficient heat energy is applied. Hence a force may be developed which is able to operate through a distance. The product of this force times the distance through which it operates gives the mechanical work done. If the substance heated is a solid body, the distance factor for a given mass of material is small. If a substance in the liquid state is used the distance factor is somewhat greater, as may be seen by reference to a table of coefficients of expansion in any engineers' handbook. Thus substances in the solid and liquid states may be utilized as agents in transforming heat energy into mechanical work, but the distance factor per unit of material is so small that very large forces must be generated in order to obtain appreciable quantities of work.

Gases and vapors expand approximately 100 times more than solids for the same temperature increments. The distance factor is much larger for gases than for liquids or solids. For this reason the expansion of gases and vapors is utilized altogether as a means for converting heat energy into mechanical work on a commercial scale. This use of gases and vapors makes a complete mathematical knowledge of their behavior under all possible changes absolutely essential to engineers.

Suppose a piston, in a cylinder filled with a gas as in Fig. 3, having an area of A square feet, is pushed forward by a uniform force of P pounds per square foot due to the gas pressure behind the piston through a distance of D feet. Then

(62)

$$F = PA,$$

and

$$(63) \quad W = PAD.$$

But

$$(64) \quad V = AD.$$

Therefore

$$(65) \quad W = PV.$$

If the equation (65) is fitted to the conditions represented in Fig. 4 it becomes

$$(66) \quad W = P_1(V_2 - V_1).$$

In the above equations the symbols represent the following quantities:

A = area of piston in sq. ft.,

D = distance through which piston moves in ft.,

F = total force acting on piston in lbs.,

P = force per sq. ft. of piston in lbs.,

V = volume displaced by piston in cu. ft.,

W = work done by one stroke of the piston in ft. lbs.

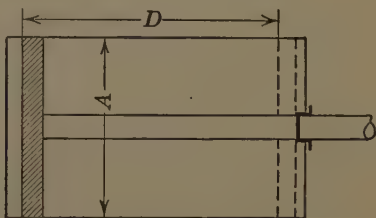


FIG. 3.—CYLINDER AND PISTON

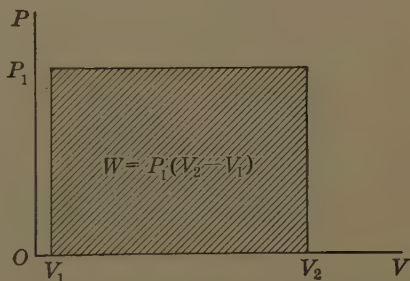


FIG. 4.—WORK AREA ON PV -PLANE

It is thus seen that an area on the PV -plane, as shown by Fig. 4, when drawn to scale, represents the mechanical work in foot-pounds done by a moving piston. It will be recalled that it was proved by integral calculus that, in general, the area of a figure on the xy -plane whether bounded by curves or straight lines is

$$(67) \quad \text{Area} = \int_a^b y dx, \quad \text{or} \quad \text{Area} = \int_a^b x dy.$$

Likewise work in foot-pounds on any PV -Plane is always

$$(68) \quad W = \int_a^b P dV, \quad \text{or} \quad W = \int_a^b V dP.$$

46. Unit in which R is Measured. It is seen by comparing equation (32), page 44, and equation (65) that R represents the foot-pounds of work done by one pound of gas when its temperature changes one degree Fahrenheit at constant pressure. *That is, R is foot-pounds of work per degree Fahrenheit for one pound of gas.* To be perfectly general the statement should be: R represents units of work per degree of temperature for unit weight of gas at constant pressure. Its value depends upon the nature of the gas and the system of fundamental units used. In the same way the molecular gas constant, 1545, is work units at constant pressure per degree of temperature for a weight of gas corresponding to the molecular weight number of the gas all in English units.

47. External and Internal Work. According to the kinetic theory, temperature rise of a substance indicates an increase in molecular activity. Both amplitude and velocity of vibration of the molecule increase causing an increase in the volume of the substance. When the state of a substance is changed from liquid to gaseous by application of heat energy there is considerable increase in volume. If the substance is subjected to external pressure, as for example the pressure of the atmosphere the change in volume takes place in opposition to that pressure acting as a force. Hence external work is done in accordance with equations (66) and (68). The work energy of such an expansion is stored not in the substance itself but in its surroundings. This work of expansion is called *external work*.

The energy used up in increasing the amplitude and velocity of molecular activity, as indicated by temperature rise, and the energy used up in separating widely the molecules from each other against the force of their mutual attraction, and in imparting motion of translation during change of state is called *internal work*. Any substance in the liquid or gaseous state possesses internal potential energy which becomes available for external use when the state of the substance is reduced from the gaseous to the liquid or from the liquid to the solid state by cooling.

The difference between external work and internal work may

perhaps be better understood if the process of heating a pound of water from 32° F. to 212° F., the boiling point at atmospheric pressure, and evaporating it into steam at the latter temperature is traced step by step. When heat is first added, the temperature begins to rise indicating greater molecular activity. A very small increase in volume also accompanies this temperature rise. These changes continue until 180 B.t.u. have been added when the temperature has become 212° F. *The heat energy absorbed in increasing the temperature only when no change of state occurs is called internal energy.* The energy that has been utilized in producing the slight increase in volume of the pound of water does *external work* against the pressure of the atmosphere. The volume of one pound of water at 32° F. is 0.01602 cu. ft. The volume at 212° F. increases to 0.01670 cu. ft. Hence the increase in volume equals 0.00068 cu. ft. The foot-pounds of external work done against a pressure of one standard atmosphere is

$$14.696 \times 144 \times 0.00068 = 1.44 \text{ ft. lbs.}$$

In B.t.u. this equals $1.44 \div 778.57 = 0.00185$ B.t.u. This number is so small that engineers usually neglect it in all heat computations.

If the addition of heat energy continues, the temperature will no longer rise, but the water begins to change over from the liquid to the gaseous state. If the pressure is held constant at that of the atmosphere, the volume change now becomes much larger. If the heat addition continues until 970.4 B.t.u. have been imparted to the boiling water, it is all converted into steam at a temperature of 212° F. The volume has increased from 0.01670 cubic feet to 26.79 cubic feet. This is about 1600 times the volume of the water in the liquid state at 212° F. The *external work* done during this change is

$$\frac{(26.79 - 0.0167) 144 \times 14.696}{778.57} = 72.77 \text{ B.t.u.}$$

The remainder of the heat energy absorbed (897.63 B.t.u.) has been used up in increasing the *internal energy* of the H_2O , and

is called *internal work*. The internal energy may be segregated into three items:

1. The average distance between the molecules has been enormously increased against the force of mutual attraction,
2. The mean velocity of the molecular vibratory movements has been increased,
3. The molecules have acquired movements of translation at high velocity.

48. Latent Heat of Evaporation. The total heat required to evaporate a pound of water at 212° F. into steam at 212° F. is called latent heat of evaporation, or simply latent heat. It is generally considered to be made up of two parts, the external work part or external latent heat and the internal work part or the internal latent heat. For every different pressure of evaporation there is a new boiling point and a new value of latent heat of evaporation. As the pressure at which evaporation takes place increases, the boiling point rises while the latent heat of evaporation decreases in value. These facts may be observed by referring to any steam table.

49. Fundamental Heat-Energy Equation. When a single substance or a system composed of several substances (as a mixture of gases or a mixture of a liquid and its vapor) receives heat energy, there are only three possible changes which may take place, viz:

1. The temperature of the substance or system may rise,
2. Internal work may be done, and, or,
3. External work may be done.

All these changes may take place simultaneously or only two of them or even only one of them may result from heat addition. There is no direct mathematical proof of these statements but experience has always been in accord with them. This state of affairs may be best expressed by the following equation in the differential form:

$$(69) \quad \pm dQ = \pm dH \pm dI \pm AdW.$$

Equation (69) is the fundamental heat equation. In the integral form it becomes

$$(70) \quad \pm \int_{Q_1}^{Q_2} dQ = \pm \int_{H_1}^{H_2} dH \pm \int_{I_1}^{I_2} dI \pm A \int_{W_1}^{W_2} dW.$$

Integrating equation (70), one obtains

$$(71) \quad \pm (Q_2 - Q_1) = \pm (H_2 - H_1) \\ \pm (I_2 - I_1) \pm A(W_2 - W_1).$$

Equation (71) may be simplified considerably if it is understood that $Q_2 - Q_1 = Q$; $(H_2 - H_1) = H$; and so on. It then becomes

$$(72) \quad \pm Q = \pm H \pm I \pm AW.$$

In the above equations,

$A = 1/J = 1/778.57$, if Q is in B.t.u.

Q = the heat energy given out by or added to a system,

H = the heat energy used in changing the temperature of the system,

I = the heat energy used in doing internal work,

AW = the heat energy used in doing external work.

The minus signs in equations (69) to (72) may be interpreted to indicate that heat energy is given out by the body or the system of bodies. The plus signs would then indicate that the heat energy is received by the system.

50. Joule's Experiment.¹ In 1845 Joule proved that the *internal work is zero when a perfect gas expands without doing any external work*. This fact is another foundation principle in the science of thermodynamics of gases. This principle was proven by a simple laboratory experiment which has become known as *Joule's experiment*.

Two copper bottles, A and B, as in Fig. 5, having equal capacities were connected by a tube and stopcock C. Bottle A contained compressed air at a pressure of 22 atmospheres, while bottle B had been evacuated as completely as possible. The

¹ See Philosophical Magazine, vol. 26, 1845, p. 369.

bottles were then submerged in a water bath D, and allowed to stand several hours until complete thermal equilibrium was established as indicated by a thermometer. Cock C was then

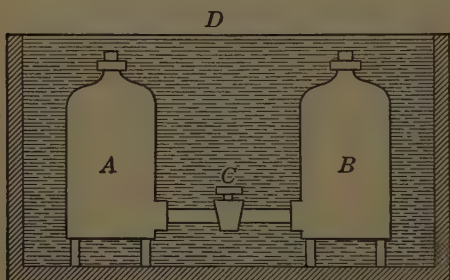


FIG. 5.—Joule's EXPERIMENT

opened allowing the air to expand into bottle B equalizing the pressure in the two bottles.

The bottles and contained air constituted a closed thermodynamic system. When the cock was opened the expanding air could do no external work since the

volume of the system before and after the expansion was the same, hence the total volume change was zero. Then the term AW of equation (72) becomes zero. Also Q was made zero since no external heat energy, as such, was added to or taken away from the system. Joule watched the thermometer in the bath for any change in temperature produced by the expanding gas. He found the temperature change to be zero. Thus H of equation (72) was proved to be zero. Since Q and AW were made equal to zero and H was measured and found to be zero, Joule proved that under those conditions I must be equal to zero for a gas. This means that when a gas is heated none of the heat energy absorbed is transformed into internal work. It is all transformed into temperature energy or into external work or it is partly transformed into temperature energy and partly into external work. Hence for a perfect gas equation (72) becomes

$$(73) \quad \pm Q = \pm H \pm AW.$$

This fact may be stated thus; when heat energy is added to or taken away from a perfect gas without change of state, there is no change in the internal work. This is known as *Joule's law*.

In 1852, Joule and Sir William Thomson (Lord Kelvin) repeated the experiment under more exacting conditions, using a porous plug in place of the cock C of Fig. 5. They observed

slight temperature changes, for some gases a rise and for others a drop in temperature. This phenomena has been named the *Joule-Thomson effect*.¹ From their experiments they concluded that a perfect gas would show no temperature change.

Equation (73) approximates the truth only so long as the substance heated remains in the gaseous state. *When changes of state occur, the complete equation as stated in (72) must be used.*

51. Heat Added. Case I. When M pounds of a perfect gas are heated at *constant pressure* from absolute temperature T_1 to absolute temperature T_2 the quantity of heat, Q , absorbed by the gas is expressed by

$$(74) \quad Q = \hat{c}_p M(T_2 - T_1).$$

For instance, if in a cylinder with a movable, weighted piston as shown in Fig. 6, M pounds of a perfect gas is heated under constant pressure it expands forcing the piston upward and external work is done due to the change in volume. At the same time heat energy is absorbed by the gas raising its temperature. The foot-pounds of external work done may be computed as follows:

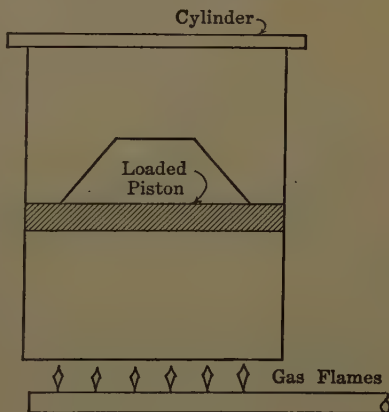


FIG. 6—HEAT ADDITION AT
CONSTANT PRESSURE

$$(75) \quad W = P \int_{v_1}^{v_2} dV$$

Substituting equation (31),
p. 44, in (75), one finds

$$(76) \quad W = MR(T_2 - T_1) \quad (\text{in foot-pounds}).$$

From Joule's law as expressed in equation (73), the heat energy used up in increasing the temperature only is found to be

$$(77) \quad H = Q - AW.$$

¹ Philosophical Transactions, vol. 143 (1853), p. 357; vol. 144 (1854), p. 321; vol. 152 (1862), p. 579.

Substituting equations (74) and (76) in (77), one obtains

$$(78) \quad H = \bar{c}_p M (T_2 - T_1) - AMR(T_2 - T_1),$$

and

$$(79) \quad H = M(\bar{c}_p - AR)(T_2 - T_1).$$

The value of H in (79) is expressed in B.t.u. The symbol $\bar{c}_p$ represents the mean specific heat of the gas under the conditions of constant pressure change. The other symbols represent quantities as explained on pages 43, 44 and 61.

52. Heat Added. Case II. If now the cylinder of Fig. 6 is again partly filled with M pounds of a perfect gas and heat energy is added, but in this instance the piston is held stationary thus keeping the volume of the gas constant during the heat change, no external work will be done. The force acting will be variable, but the distance through which it acts is zero, hence the *external work is zero*. Then all the heat energy absorbed by the gas is used to *increase its temperature*. This change may be represented by the symbols used in equations (78) and (79) except $\bar{c}_v$ which represents the mean specific heat of the gas during constant volume charge. Thus

$$(80) \quad H = Q = \bar{c}_v M (T_2 - T_1).$$

It is seen by equations (79) and (80) that there are *two kinds of specific heat* for a gas, depending upon the conditions under which it is heated. The value of $\bar{c}_p$ must be the larger since not only is the temperature increased but external work is done at the same time. In the case of $\bar{c}_v$ no external work is done, and its value is smaller.

If for these two cases of heat change, the values of M , T_1 , and T_2 are made the same, equations (79) and (80) may be equated thus

$$(81) \quad M(\bar{c}_p - AR)(T_2 - T_1) = \bar{c}_v M (T_2 - T_1),$$

$$(82) \quad \bar{c}_p - AR = \bar{c}_v$$

and

$$(83) \quad \bar{c}_p - \bar{c}_v = AR,$$

and

$$(84) \quad \frac{\bar{c}_p}{A} - \frac{\bar{c}_v}{A} = R.$$

Since Joule's equivalent equals

$$\frac{1}{A} = J,$$

if one sets

$$\bar{c}_p J = j_p,$$

$$\bar{c}_v J = j_v,$$

equation (84) becomes

$$(85) \quad j_p - j_v = R.$$

In equation (85) j_p and j_v represent the respective specific heats measured in foot-pound units.

Again, it is seen that R , for a gas, is the difference between the two specific heats of that gas when expressed in foot-pounds units. The complete definition for R as shown by equation (85) is; the value of R for a given gas, in English units, is the number of foot-pounds of external work done when one pound of that gas expands at constant pressure while its temperature is increased one degree Fahrenheit. The value of R may likewise be stated in c.g.s. units, but a different number must be used to express its value.

The kinetic theory of gases indicates that the values of c_p and c_v for a perfect gas would be constants at all temperatures. In that case, the mean and instantaneous values of specific heat would be constant. That is

$$(86) \quad c_p = \bar{c}_p \quad \text{and} \quad c_v = \bar{c}_v.$$

53. Ratio of Specific Heats. The ratio of the two specific heats for a perfect gas is

$$(87) \quad \frac{c_p}{c_v} = \frac{\bar{c}_p}{\bar{c}_v} = k.$$

Whether the instantaneous values or the mean values of specific heat are employed their ratio is always represented by the symbol

k as in equation (87). Since for any gas found in nature the specific heat values are not constant (they increase with temperature), the value of k is not a constant. Its value generally decreases as the temperature rises.

The following relations between c_p , c_v , R and k are useful and important. Equation (83) may be transformed as follows:

$$(88) \quad \frac{c_p}{c_v} - 1 = \frac{AR}{c_v},$$

$$(89) \quad k - 1 = \frac{AR}{c_v},$$

$$(90) \quad c_v = \frac{AR}{k - 1},$$

and

$$(91) \quad \frac{c_v}{A} = j_v = \frac{R}{k - 1}.$$

In a similar manner from equations (82), (83), (84), and (87) expressions for c_p and j_p may be obtained, as follows:

$$(92) \quad c_p = \frac{ARk}{k - 1},$$

and

$$(93) \quad j_p = \frac{Rk}{k - 1}.$$

By introducing the value of R in terms of the molecular gas constant, equations (91) and (93) become

$$(94) \quad j_v = \frac{1545}{m(k - 1)},$$

$$(95) \quad j_p = \frac{1545k}{m(k - 1)}.$$

Table 8 contains the values of the constants of the more important gases in which engineers are most frequently interested.

TABLE 8
PHYSICAL CONSTANTS OF GASES

Gas	Molecular Weight	Cu. Ft. per Lb. at 32° F. and Atmos. Press	R	c_p	c_v	k
Air.....	28.96	12.39	53.36	0.23973	0.17083	1.4034
Ammonia.....	17.03	20.80	90.73	0.5132	0.3918	1.310
Argon.....	39.88	9.00	38.75	0.1246	0.0747	1.667
Carbon dioxide	44.00	8.15	35.12	0.1998	0.1534	1.302
Carbon monoxide	28.00	12.81	55.19	0.2478	0.1764	1.405
Helium.....	4.00	89.70	386.31	1.2425	0.7540	1.667
Hydrogen.....	2.02	177.96	766.48	3.4028	2.4160	1.408
Methane.....	16.04	22.37	96.34	0.5294	0.4040	1.310
Nitrogen.....	28.02	12.81	55.15	0.2471	0.1759	1.405
Oxygen.....	32.00	11.21	48.29	0.2200	0.1575	1.397

The items in Table 8 are computed for conditions and in the manner as explained below.

1. All values varying with temperature and pressure are for a temperature of 32° F. and a pressure of one standard atmosphere of 14.696 pounds per square inch.

2. The molecular weights are based upon the values established by the International Committee on Elements and Atomic Weights in 1925.

3. The weight of one liter of pure dry air containing 0.04 per cent of CO₂ and measured at 32° F. and at a pressure of 14.696 pounds per square inch is taken as 1.2256 grams. Hence one cubic foot of pure dry air weighs 0.080712 pounds. The volume of one pound of air at the conditions of temperature and pressure specified is 12.39 cubic feet.¹

4. The specific gravity of oxygen (air = 1.000), under the conditions mentioned in 2, is 1.1050. Therefore one cubic foot of oxygen weighs 0.0891868 pounds, and its specific volume is 11.212 cubic feet at the specified conditions of temperature and pressure.

5. The molecular gas constant for oxygen is then

$$(96) \quad mR = \frac{Pvm}{T}.$$

¹ See *Circular of the U. S. Bureau of Standards No. 19.*

Numerically, the right side of (96) is equal to

$$(97) \quad \frac{2116.3 \times 11.212 \times 32}{491.6} = 1545.2$$

6. The molecular gas volume for oxygen is

$$(98) \quad mv = 32.00 \times 11.212 = 358.78 \text{ cu. ft.}$$

This is a constant for all gases. It is known as the volume of one *mol*.

7. The value of Joule's mechanical equivalent is accepted as being 778.57.

8. The values of c_p and k used are taken, by permission, from Partington and Shilling.¹

9. The following relations are assumed to be strictly fulfilled for each of the gases as would be the case for a perfect gas:

$$(99) \quad R_m = \frac{1545.2}{m},$$

$$(100) \quad \frac{c_p}{c_v} = \frac{\bar{c}}{\bar{c}_v} = k = \text{a constant.}$$

Then c_v is computed from (100) as follows:

$$(101) \quad c_v = \frac{c_p}{k}.$$

Since gases such as air, oxygen, helium, hydrogen, and nitrogen are not perfect gases, the relations expressed by equations (32), (85), (99), and (100) are not exactly fulfilled for them. For example, take the generally accepted values of c_p and c_v from Table 8, and the accepted value of J , (778.57), to determine the value of R for one pound of air by equation (85); since

$$(102) \quad j_p = Jc_p$$

and

$$(103) \quad j_v = Jc_v,$$

therefore equation (85) becomes

$$(104) \quad R = J(c_p - c_v) = 778.57(.23973 - .17083) = 53.64.$$

¹ See J. R. Partington and W. G. Shilling, *The Specific Heats of Gases*, p. 201. E. Benn, Ltd., London, 1924.

By comparing the value of R for air as shown in equations (33), (104), and in Table 8, it is seen that there is not exact agreement. This discrepancy is due probably to slight errors in laboratory measurements as well as to the fact that air is not a perfect gas. Corresponding discrepancies will be found for all gases in nature.

54. Complete Specifications of a Perfect Gas. The complete specifications of a perfect gas may now be stated as follows:

1. Boyle's law is strictly fulfilled for all pressures and volumes.
2. Charles' law is strictly true at all temperatures.
3. As a consequence of specifications 1 and 2 the equation of state (32) is always exactly fulfilled.
4. The Joule-Thomson effect is zero.
5. The specific heats (c_p and c_v) are fixed and constant in value at all pressures, temperatures, and volumes.
6. The value of the ratio of specific heats, k , is a constant.
7. The difference between the two specific heats is a constant.
8. The internal energy of the gas depends upon its temperature only. It is independent of the pressure and volume.

A perfect gas has never yet been found in nature. It probably does not exist. But the so-called permanent gases such as helium, hydrogen, nitrogen, oxygen, air, carbon dioxide, and others in the same class are commonly treated as if they were perfect in the above defined respects. Hence all the statements made and all equations derived for perfect gases are only approximately true for the actual gases found in nature.

55. Van der Waals' Equation.¹ Nearly sixty variations in the form of the equation of state have been proposed by as many investigators, to account for the deviations of the permanent gases from the exact relation $PV = RT$. Some of the proposed equations are complicated and inconvenient for computation purposes. One of the most satisfactory equations in many respects was proposed by Johannes D. Van der Waals in his doctor's dissertation at the University of Leyden, in 1873.

¹ See Partington and Shilling, *The Specific Heats of Gases*, p. 29.

Although Van der Waals' equation is not so accurate, especially for higher pressures, as several others, it is much quoted because it gives a very clear picture of the physical relations existing within a gas. Van der Waals' equation is

$$(105) \quad \left(P + \frac{a}{v^2} \right) (v - b) = RT.$$

In (105) a and b are constants differing for each gas under consideration.

As the pressure of a gas is increased, the molecules are crowded closer together, and the mutual forces of cohesion between the molecules due to this squeezing together of their masses increase. This tends to reduce the outward pressure effects which are measurable with instruments. The pressure, P , as measured should be the full value unaffected by the gradually increasing attractive forces as the molecules are forced closer and closer together by the rising pressure. Van der Waals assumed this small pressure correction to be equal to a constant, a , divided by the square of the volume.

The constant b represents the final volume which would be occupied by the molecules if they were compressed solidly together into a compact body. It is known that solids and liquids are rather spongy materials capable of considerable reduction in volume, if sufficiently high pressures could be applied, before the molecules would be forced into actual contact. The symbol b is sometimes referred to as the covolume. Then $(v - b)$ represents the vacant space contained within an enclosure in which molecules of a substance are floating about more or less distant from each other. Van der Waals' equation indicates that it is only this unoccupied space which changes with change in pressure according to Boyle's Law.

Equation (105) may be transformed into a cubic in v as follows:

$$(106) \quad Pv^3 - (bP + RT)v^2 + av - ab = 0.$$

if a and b in (106) are considered to be constants, as did Van der Waals, it has one real root and two imaginary ones; or it has three real roots all equal. If the later be considered the case, the values of a and b are found to be as follows:

$$(107) \quad a = \frac{27R^2T^2}{64P},$$

and

$$(108) \quad b = \frac{RT}{8P}.$$

The values of a and b in the English system of units for three of the commonest gases are as shown in Table 9.¹

TABLE 9
CONSTANTS FOR VAN DER WAALS' EQUATION

Gas	a	b
Air	877.07	0.02032
Oxygen.....	721.24	0.01595
Nitrogen	931.60	0.02207

Van der Waals' equation is not satisfactory over a wide range of pressure, especially toward the higher values, because the constant a is not really a constant quantity. Berthelot² proposed equation (109) which is in some respects much better than Van der Waals.'

$$(109) \quad P + \frac{a}{Tv^2} (v - b) = RT,$$

where a , b , and R are constants; but a here has a different value from a in Van der Waals' equation.

Not one of all the half-hundred or more equations which have been proposed from time to time is perfectly satisfactory for all three states of matter. Such an ideal equation will probably never be found. So far as the engineer, dealing with the problems of gas engines and steam engines, is concerned it is not important that such an equation be found anyhow. If such an equation were available it would be too complicated for easy and rapid solution; hence resort to vapor tables, as is now the practice, would still be necessary.

56. Equations for the Instantaneous Values of Specific Heats of Gases. Since there are no perfect gases, the values of the specific heat at constant pressure and the specific heat at constant volume are not constants. These variable quantities have been found to be represented fairly well by empirical equations as given in Table 10. These equations are not the final answer but are in accord with the best and most authoritative information available at present.

¹ Values of a and b for other gases may be obtained from G. W. C. Kaye and T. H. Laby, *Tables of Physical and Chemical Constants*, p. 34.

² Archives Neerlandaises des Sciences Exactes et Naturelles, vol. 5, 1900, p. 417.

TABLE 10
INSTANTANEOUS SPECIFIC HEATS OF GASES

GAS	EQUATIONS FOR VALUES OF T BETWEEN 492° F. AND 2500° F. AND FOR ATMOSPHERIC PRESSURE
Air *	$c_v = 0.1679 + 5.56 \times 10^{-6}T + 5.97 \times 10^{-10}T^2$ $c_p = 0.2368 + 5.28 \times 10^{-6}T + 6.70 \times 10^{-10}T^2$
CO ₂ †	$c_v = 0.1269 + 5.36 \times 10^{-6}T - 5.02 \times 10^{-9}T^2$ $c_p = 0.1747 + 5.04 \times 10^{-6}T - 3.79 \times 10^{-9}T^2$
Nitrogen *	$c_v = 0.1720 + 6.54 \times 10^{-6}T + 5.16 \times 10^{-10}T^2$ $c_p = 0.2433 + 6.28 \times 10^{-6}T + 5.84 \times 10^{-10}T^2$
Oxygen *	$c_v = 0.1557 + 3.65 \times 10^{-6}T + 5.31 \times 10^{-10}T^2$ $c_p = 0.2181 + 3.26 \times 10^{-6}T + 6.46 \times 10^{-10}T^2$
Steam (H ₂ O) †	$c_v = 0.576 - 4.75 \times 10^{-4}T + 2.72 \times 10^{-7}T^2 - 3.82 \times 10^{-11}T^3$ $c_p = 0.74 - 5.70 \times 10^{-4}T + 3.30 \times 10^{-7}T^2 - 4.83 \times 10^{-11}T^3$

T = degrees F. abs.

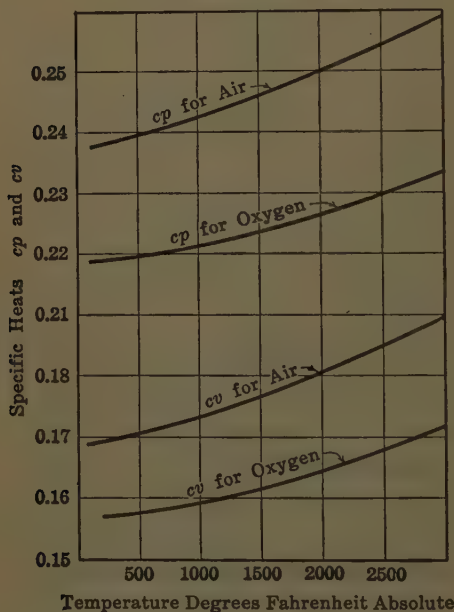


FIG. 7.—INSTANTANEOUS SPECIFIC HEATS OF AIR AND OXYGEN

* Computed from equations in J. R. Partington and W. G. Shilling's *The Specific Heats of Gases*, p. 145. E. Benn, Ltd., London.

† Computed from unpublished equations kindly furnished by W. G. Shilling.

When the equations of Table 10 are plotted as has been done for air and oxygen in Fig. 7, it is seen that they have a slightly upward curvature as the temperature increases, showing that the specific heat values rise faster than the simple straight line law would indicate.

Physicists and chemists usually employ equations which give specific heat values in calories per gram molecule. Molecular specific heat values are then

numbers which show the quantity of heat in calories required to raise the molecular weight in grams of a substance one degree Centigrade. This weight is called a *mol* or *gram-mol*. (Some writers use the spelling *mole*.) In the same way, B.t.u. per pound-mol may be used. The equation for c_v for nitrogen expressed in terms of the gram-mol is

$$(110) \quad C_v = 4.820 + 3.30 \times 10^{-4}T_c + 4.70 \times 10^{-8}T_c^2,$$

where

T_c = absolute temperature in degrees Centigrade,

C_v = molecular specific heat.

PROBLEMS

1. In Table 10, the temperatures used for computing c_p and c_v must be expressed in degrees Fahrenheit absolute. (a) Change them for CO_2 so that degrees as read from the Centigrade thermometer may be used. (b) Change them for air so that absolute Centigrade degrees may be used. (c) Change them for air so that direct readings of the Fahrenheit thermometer may be used.

2. Compute the values of mean specific heats c_p and c_v for CO_2 and for air between 70°F. and 2000°F.

3. The value of k for a given gas is 1.667, the value of $R = 386.3$. Find the values of c_p and c_v . (Use equations (88) to (93).)

4. Using equation (31) and others, and the following data, compute for each case the final pressure, final volume, B.t.u. added, the change of internal energy in B.t.u., and the external work done when fifteen pounds of air are heated from 60°F. to 500°F. : (a) At constant pressure; (b) at constant volume (assume c_v constant at 0.171).

5. Repeat Problem 4 but use Van der Waals' equation of state (equation (105), page 70).

6. Compute the volume of a gram-mol in liters for gases.

7. Change the constants in Table 10 to give values of the specific heats in terms of the pound-mol.

CHAPTER VIII

GAS EXPANSIONS

57. Property Planes. Experience has taught that very valuable and important information upon heat energy and work relations may be obtained by studying the relations between the cardinal properties as gases are heated with or without expansion. Combining the five cardinal properties of gases, any two at a time, will give ten possible combinations. These ten pairs of properties may be used as the coördinate axes of ten property planes upon which curves may be drawn. The most useful combinations to engineers are two in number, viz: PV and TN where P represents pressure, V volume, T temperature, and N entropy.

When one of a pair of properties of a gas is changed progressively through a series of values, the other property also passes through a mathematically related series of values. By plotting the values of the ten pairs of properties on rectangular coördinates, curves on the ten property planes are produced. For example, by plotting the values of pressure and volume for a given weight of a gas through a series of related values on the PV -plane and connecting the points by a continuous line the PV -curve is obtained. (See Cardinal Properties, p. 37 above). The ten possible property planes are indicated in Table 11.

TABLE 11
PROPERTY PLANES

PROPERTIES	PROPERTY PLANES	PROPERTIES	PROPERTY PLANES
P	$PV, PT, PN, PE_i,$	N	and NE .
V	$VT, VN, VE_i,$	E_i	
T	TN, TE_i		

By properly controlling the changes between P and V , the PV -curve may be made a straight horizontal line. Then P is

constant throughout the change. Or the curve may be made a straight vertical line by keeping the volume constant during heat addition. Also by proper control, curves of every gradation between may be obtained as illustrated in Figure 8. All these PV -curves may be represented by the general equation

$$(111) \quad PV^n = K,$$

where

K = a constant,

n = a number between zero and plus infinity,

P = pressure in lbs. per sq. ft.,

V = volume of gas in cu. ft.

Equation (111) is the equation of a line on the PV -plane. Equations of state (31), (32), or (105) are equations of a point on the PV -plane. The point may lie on the line or curve whose equation is (111). In fact all the points in the curve whose equation is (111) may be represented, one by one, by (31), (32), or (105).

The value of the exponent n in equation (111) may be any number between zero and plus infinity. But it must remain a constant throughout any one curve. When $n = 0$ the curve is a horizontal straight line. When n equals plus infinity the curve is a straight vertical line. The value of n is determined by the schedule of heat addition or extraction and upon the amount of external work done during the pressure-volume changes.

58. Expansion and Compression. If the value of V is increasing during a change, it is called *expansion*. Generally, the value of P decreases during expansion. If the value of V is decreasing as the change progresses, the change is called *compression*. The pressure generally increases in value during compression. But it would be possible to schedule a series of heat and work changes during which there would be compression with falling pressure.

59. Special Changes for Gases. There are five special changes (one for each cardinal property of a gas) which are of special

interest and importance due to the fact that each of the cardinal properties, in turn, is constant in value throughout the change under consideration. These changes are given distinctive names which indicate the cardinal property remaining constant. The changes are as follows:

1. *An isothermal change* is one during which heat is added and external work performed while the temperature remains constant throughout. The path or curve on any of the ten property planes which represents the isothermal change is called an isothermal curve or simply an isothermal.

2. During *an isometric change* the volume remains constant. The path or curve on any property plane which represents an isometric change is called an isometric curve.

3. During *an isobaric (or isopiestic) change* the pressure is constant. The path representing an isobaric on any property plane is called an isobaric or constant pressure curve.

4. When the internal energy of a gas remains constant the change is called *an isodynamic change* and the corresponding path on any property plane is an isodynamic curve.

5. A change during which no heat energy is received or given up by the gas is called *an isentropic or adiabatic change*. If any external work is done during this change it must be at the expense

of the internal energy already stored in the gas before the change starts. The path representing this change on any property plane is called an adiabatic curve or simply an adiabatic.

In Fig. 8, the curve for $n = 0$ is an *isobaric*. The curve for $n = 1.00$ or

$$(112) \quad PV = C$$

is *an isothermal*. The curve for $n = k$ or

$$(113) \quad PV^k = K_1$$

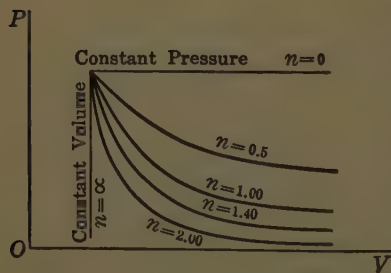


FIG. 8.—POLYTOPICS ON THE PV -PLANE

is *an adiabatic*. Of the five special changes, these two whose paths on the PV -plane are represented by equations (112) and (113) are the most important, for the engineer. That equation (113) represents the path of an adiabatic change will be proven a little later. In Fig. 8, the vertical straight line for $n = +\infty$ is an isometric curve.

The whole family of curves represented by the general equation (113) is given the generic term *polytropic curves*. But when any particular curve of this family is to be studied it is given the special name, as explained above, whenever possible.

60. External Work Done during a Polytropic Change. Suppose a quantity of gas expands along a polytropic curve from V_1 to V_2 represented on the PV -plane by the polytropic equation

$$(114) \quad PV^n = P_1 V_1^n = P_2 V_2^n = K,$$

as illustrated in Fig. 9. By equation (67), the work done is

$$(115) \quad W = \int_{V_1}^{V_2} P dV.$$

From equation (114), the value of P is

$$(116) \quad P = \frac{K}{V^n}.$$

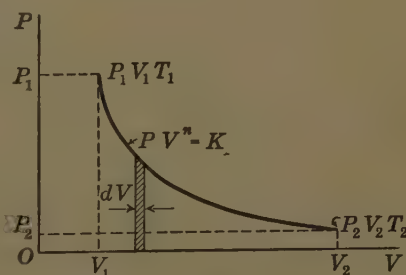


FIG. 9—WORK UNDER A POLYTROPIC ON THE PV -PLANE

Hence (115) becomes

$$(117) \quad W = \int_{V_1}^{V_2} K \frac{dV}{V^n} = K \int_{V_1}^{V_2} V^{-n} dV.$$

When integrated, (117) becomes

$$(118) \quad W = \frac{P_1 V_1 - P_2 V_2}{n - 1} \text{ in ft.-lbs.}$$

But, from the equation of state (31),

$$(119) \quad P_1 V_1 = MRT_1 \quad \text{and} \quad P_2 V_2 = MRT_2.$$

Substituting (119) in (118), one finds

$$(120) \quad W = \frac{MR(T_1 - T_2)}{n - 1} \text{ in ft.-lbs.}$$

It must be remembered that the equations in this section, as in subsequent sections, are based upon the assumption that the *expanding medium is a perfect gas*.

61. External Work Done during an Adiabatic Change. During an adiabatic change there is no flow of energy in the form of heat into or away from the changing medium. Hence, in equation (73), page 62, the value of Q is zero, and equation (73) may be written

$$(121) \quad AW = -H.$$

From equation (80) it is seen that

$$(122) \quad H = \bar{c}_v M (T_1 - T_2).$$

The value of W , as in (120), may be obtained from (121) and (122); it is

$$(123) \quad W = \frac{\bar{c}_v M (T_1 - T_2)}{A}.$$

By equation (91) page 66, (123) becomes

$$(124) \quad W = \frac{MR(T_1 - T_2)}{k - 1} \text{ in ft.-lbs.}$$

Comparing equations (120) and (124), one sees readily that they would be identical if

$$(125) \quad n = k.$$

Hence it may be concluded that for an adiabatic change the value of n equals k , the ratio of the specific heats; and the equation representing *an adiabatic change* is thus proved to be

$$(126) \quad PV^k = P_1 V_1^k = P_2 V_2^k = K_1.$$

Also by taking steps similar to equations (115) to (118) inclusive,

and referring to Fig. 9, one may show that

$$(127) \quad W = \frac{P_1 V_1 - P_2 V_2}{k - 1} \text{ in ft.-lbs.}$$

62. External Work Done during an Isothermal Change. (Refer to Fig. 10.) During isothermal change, the temperature is constant and Boyle's law represents the relation between pressure and volume. Thus the equation for isothermal expansion is

$$(128) \quad PV = P_1 V_1 \\ = P_2 V_2 = C.$$

Proceeding as in equations (114) to (118) inclusive, one obtains an expression for the external work done in the form

$$(129) \quad W = \int_{V_1}^{V_2} P dV,$$

But equation (128) gives

$$(130) \quad P = \frac{C}{V}.$$

Hence (129) becomes

$$(131) \quad W = C \int_{V_1}^{V_2} \frac{dV}{V}.$$

Integrating (131), one obtains

$$(132) \quad W = C (\log_e V_2 - \log_e V_1) = C \log_e \frac{V_2}{V_1}$$

where the value of W is expressed in foot-pounds. The following relation may be obtained from equation (128):

$$(133) \quad \frac{V_2}{V_1} = \frac{P_1}{P_2} = r.$$

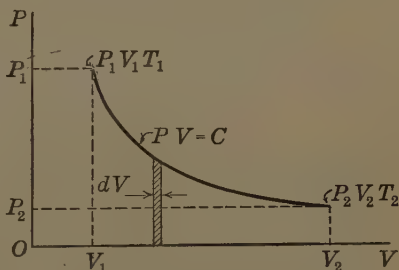


FIG. 10.—WORK UNDER THE ISOTHERMAL

The quantity r is called the *ratio of expansion*. It always has a value greater than one. Using equations (128) and (133), one may express the value of W in (132) as follows:

$$(134) \quad W = P_1 V_1 \log_e r = P_2 V_2 \log_e r,$$

and

$$(135) \quad W = MRT_1 \log_e r = MRT_2 \log_e r.$$

But for isothermal expansion the temperature is constant, hence equation (135) becomes simply

$$(136) \quad W = MRT_1 \log_e r.$$

In all the above equations W is always expressed in foot-pound units.

63. Relations between P , V , and T during Polytropic Changes. Equation (114) may be solved in the following manner:

$$(137) \quad \frac{P_1}{P_2} = \left(\frac{V_2}{V_1} \right)^n.$$

Or again

$$(138) \quad \frac{V_1}{V_2} = \left(\frac{P_2}{P_1} \right)^{1/n}.$$

The following equation may be obtained from (27), page 43:

$$(139) \quad \frac{T_1}{T_2} = \frac{P_1 V_1}{P_2 V_2}.$$

Equation (139) may be reduced to one in terms of T and P by substitution from (138) as follows:

$$(140) \quad \frac{T_1}{T_2} = \frac{P_1}{P_2} \left(\frac{P_2}{P_1} \right)^{1/n} = \frac{P_1}{P_2} \left(\frac{P_1}{P_2} \right)^{-(1/n)} = \left(\frac{P_1}{P_2} \right)^{(n-1)/n},$$

or

$$(141) \quad \frac{P_1}{P_2} = \left(\frac{T_1}{T_2} \right)^{n/(n-1)}.$$

Substituting (137) in (141), one finds the result

$$(142) \quad \frac{V_1}{V_2} = \left(\frac{T_2}{T_1} \right)^{1/(n-1)}.$$

By rearrangement, equations (137) to (142) may be stated in the forms shown in Table 12.

TABLE 12

RELATIONS BETWEEN P , V , AND T DURING POLYTROPIC CHANGES

$$\begin{aligned} -\frac{P_1}{P_2} &= \left(\frac{V_2}{V_1}\right)^n = \left(\frac{T_1}{T_2}\right)^{n/n-1} \\ \frac{V_1}{V_2} &= \left(\frac{P_2}{P_1}\right)^{1/n} = \left(\frac{T_2}{T_1}\right)^{1/n-1} \\ \frac{T_1}{T_2} &= \left(\frac{P_1}{P_2}\right)^{n-1/n} = \left(\frac{V_2}{V_1}\right)^{n-1} \end{aligned}$$

The relations given in Table 12 are used very frequently in thermodynamic problems. They should be memorized as thoroughly as the multiplication table.

64. Heat Changes during Expansion of a Perfect Gas. There are five cases under which a gas may expand with heat change. To find the quantity of heat involved (heat received in some cases, heat lost by the gas in other) requires special mathematical treatment for four of the cases. Joule's law as expressed in equation (73), p. 62, is

$$Q = H + AW$$

and it holds for each case, provided the gas is considered a perfect one.

Case I. When the expansion is *adiabatic* the external work is done at the expense of the heat energy already stored in the gas and no heat is added or given out; hence Q in equation (73) is zero and

$$(143) \quad H = -AW.$$

The minus sign in (143) merely indicates that the work is done at the expense of the internal heat already in the gas. It has been shown that for adiabatic expansion the exponent of V in the equation for polytropics is equal to k (see § 61).

General Case. When the exponent n does not equal k , the value of Q is not zero. In this case the heat change is equal to

the difference between H and AW thus:

$$(144) \quad Q = H - AW,$$

where

$$H = \bar{c}_v M (T_2 - T_1).$$

Substituting this value of H and the expression for AW from equation (118) in (144), one obtains the result

$$(145) \quad Q = \bar{c}_m M (T_2 - T_1) - \frac{A(P_1 V_1 - P_2 V_2)}{n - 1}.$$

If in equation (145) n is less than k , part of the heat required to perform the external work must be supplied from a source outside the expanding gas; for in that case the last term of equation (145) is greater than

$$\frac{A(P_1 V_1 - P_2 V_2)}{k - 1}$$

which represents the value of the external work in B.t.u. if the expansion had been adiabatic. (See Fig. 11.) In case n is

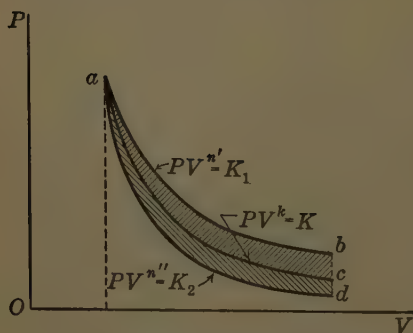


FIG. 11.—EXPANSION WITH HEAT RECEIVED OR REJECTED

greater than k the reverse is true regarding the two terms and some heat energy has to be extracted from the expanding gas in order that the path of the expansion may follow the designated law stated by

$$(146) \quad PV^{n''} = K_2.$$

The amount of heat so added or extracted may be computed in terms of A , n ,

k , P , and V using equation (145) for this purpose. From the equation of state (31), the following is obtained:

$$(147) \quad MT_1 = \frac{P_1 V_1}{R} \quad \text{and} \quad MT_2 = \frac{P_2 V_2}{R}.$$

When (147) is substituted in (145), it becomes

$$(148) \quad Q = \frac{\bar{c}_v}{R} (P_1 V_1 - P_2 V_2) - \frac{A}{n-1} (P_1 V_1 - P_2 V_2),$$

and since

$$\frac{\bar{c}_v}{R} = \frac{A}{k-1}$$

equation (148) becomes

$$(149) \quad Q = \frac{A}{k-1} (P_1 V_1 - P_2 V_2) - \frac{A}{n-1} (P_1 V_1 - P_2 V_2),$$

and

$$(150) \quad Q = A \left(\frac{1}{k-1} - \frac{1}{n-1} \right) (P_1 V_1 - P_2 V_2),$$

or

$$(151) \quad Q = A \left(\frac{n-k}{k-1} \right) \left(\frac{P_1 V_1 - P_2 V_2}{n-1} \right).$$

When n is less than k , the value of Q comes out with the negative sign indicating that there is not sufficient heat in the gas to carry out the specified *expansion*. Hence Q heat units must be supplied to the gas from an external source. If Q comes out with the plus sign which would be the case when the value of the exponent n for the expansion is greater than k , then the B.t.u. must be taken away from the gas during the expansion.

If the change were a *compression* the reverse statement as to heat added as indicated by the sign would be true since polytropic curves for an expansion and a compression intersect and cross on the adiabatic. Above the intersection (at higher pressures) would be a compression and below with decreasing pressures the change would be an expansion as indicated in Fig. 11.

Case 3. This case is for *expansion at constant pressure*, that is, $n = 0$. For this program equation (73), p. 62, holds. From equations (31) and (73) the expression for Q_p at constant pressure becomes

$$(152) \quad Q_p = \frac{\bar{c}_p}{R} P_1 (V_2 - V_1).$$

But, by equation (92),

$$(153) \quad \frac{\bar{c}_p}{R} = \frac{AK}{k-1};$$

hence (152) becomes

$$(154) \quad Q_p = \frac{AkP_1(V_2 - V_1)}{k-1}.$$

Equation (154) gives the quantity of heat which must be received by the gas in expanding from volume V_1 to volume V_2 if the change is to take place at constant pressure.

Case 4. This case is for the condition that the volume remains constant, that is the value of the exponent n is plus infinity. Since V_2 equals V_1 , the external work must be zero (see equation (66), p. 57). For this case equation (80), p. 64, is applicable as it stands or it may be transformed as follows by use of equation (31) to give the heat at constant volume, Q_v .

$$(155) \quad Q_v = \frac{\bar{c}_v V_1}{R} (P_2 - P_1).$$

But equation (90) gives

$$(156) \quad \frac{\bar{c}_v}{R} = \frac{A}{k-1}.$$

Hence equation (155) becomes

$$(157) \quad Q_v = \frac{AV_1(P_2 - P_1)}{k-1}.$$

Equations (80), (155), or (157) may be used to compute the number of B.t.u. which must be imparted to a volume of gas, V_1 , to raise its pressure from P_1 to P_2 pounds per square foot without volume increase.

Case 5. (See Fig. 10.) Under this case is considered the condition of expansion at constant temperature, that is, isothermal expansion. Since there is no temperature change the internal energy of a perfect gas would remain constant. Therefore just sufficient heat energy, Q_t , must be imparted to the gas to do the external work; hence

$$(158) \quad Q_t = AW.$$

But equation (134) gives the value of W ; hence

$$(159) \quad Q_1 = A(P_1 V_1 \log_e r) = A(P_2 V_2 \log_e r).$$

65. Thermodynamic Cycle. A series of paths which form a closed, finite area on any property plane constitutes a thermodynamic cycle. In a closed cycle, the values of the cardinal properties of a substance change through a succession of values, but always return to their initial value upon the completion of the cycle when the substance has returned to its initial state.

In any cycle, the net external work done is represented by an area on the PV -plane. Let the area $abcd$, Fig. 12, be any cycle. Then, by equations (64) and (67), the work done along the path abc is represented by the area $abche$. Along the path cda the work is negative, and is represented by the area $chead$. The net positive work is then equivalent to the difference between these two areas or to the area $abcd$.

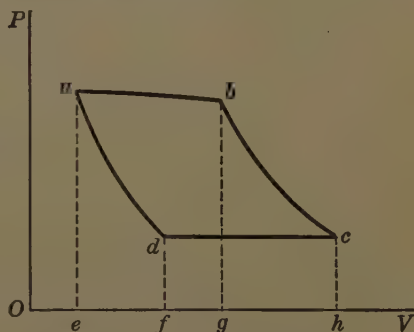


FIG. 12.—A CLOSED CYCLE

Cycles may also be represented by areas upon other property planes. For instance, the boundary lines of areas on the TN -plane represent heat cycles.

Since, by the first law of thermodynamics, work and heat are mutually convertible on an equivalent basis, we may say that areas on the PV -plane also represent heat quantities. The quantity of work done and the value of the equivalent heat energy must depend upon the kind and extent of the paths which joined together make up the cycle. But the values of the cardinal properties depend only on the state of the substance from instant to instant, and are independent of the path. Hence when the substance is brought back to its initial state at the close of a cycle the cardinal properties again possess their initial values.

If on the PV -plane the pressure units are pounds per square foot and volume units are cubic feet, the work area represents foot-pounds. Figure 12 represents the basis upon which the horsepower developed in any piston engine is computed. If the cycle be traced in the opposite direction as $adcb$, the net work is negative, or the work is done upon the substance increasing its pressure, and adding heat to it as in the air compressor.

PROBLEMS

1. Supply all missing steps in deriving equation (118) from (117), page 77.

2. Ten pounds of air expand from a pressure of 400 lbs. per sq. in. to 15 lbs. per sq. in. During the expansion the temperature is constant at 70° F. Plot on cross-section paper the PV -curve representing this change. Give all necessary computations. (It is sufficient to compute and plot five or six points for the curve. Use for scales 10 000 lbs. per sq. ft. per inch of ordinate and 10 cu. ft. per inch of abscissa.)¹

3. Plot a PV -curve for which all conditions are the same as in Problem 2, except the temperature is constant at 400° F. Plot on the same sheet with curve for Problem 2.

4. Plot the following polytropic curves on the PV -plane for 25 lbs. of air (a) when $n = 0.7$; (b) $n = 1.00$; (c) $n = k$; (d) $n = 2.2$; (e) $n = 5.0$. For each case, when $P = 5000$ lbs. per sq. ft., $t = 150^{\circ}$ F. Five points will be sufficient for each curve. Indicate all computations.

5. Given for air $P_1 = 50\ 000$; $V_1 = 8.3$; $T_1 = 950^{\circ}$ F. abs.; $P_2 = 2116.2$. Compute V_2 and T_2 when n equals 0; 0.5; 1.00; 1.25; 1.403; and 2.00.

6. Explain why one is able to "blow both hot and cold with the same breath" like the man in Aesop's Fables.

7. Air expands along the curve $PV^{1.35} = K$; $M = 25$; $p_1 = 150$ lbs. per sq. in.; $t_1 = 200^{\circ}$ F., $p_2 = 15$ lbs. per sq. in. Compute (a) foot-pounds of work represented by the area between the curve and the V -axis; (b) foot-pounds of work represented by the area between the curve and the P -axis; (c) final temperature and volume; (d) indicate whether heat is added or extracted from the air during the expansion and how much in B.t.u.

8. Repeat Problem 7 when $n = 1.00$; 1.403; and 2.00.

9. How many foot-pounds of energy must be added to 5 pounds of air to double its initial volume at a constant pressure of 50 lbs. per sq. in. abs. when the initial temperature is 100° F. What is the final temperature?

10. How many foot-pounds of energy must be added to 5 pounds of air to double its initial pressure at constant volume. When the initial temperature is 100° F., what is the final temperature?

11. How many B.t.u. must be expended upon 5 lbs. of air to compress it to one-fourth its initial volume along the curve $PV^{1.2} = K$? The initial

¹ NOTE.—In general when plotting curves on cross-section paper, divided decimally, use for scales only the numbers 1, 2, or 5, and multiples or submultiples thereof. This will insure a scale which may be read conveniently and quickly.

temperature and pressure are respectively 70° F. and 14.7 lbs. per sq. in. What would be the final temperature and pressure?

12. Twenty pounds of air expands at a constant pressure of 100 lbs. per sq. in. (and $t_1 = 75^{\circ}$ F.) to 4 times the initial volume. (a) How many B.t.u. were supplied? (b) How many foot-pounds of external work done? (c) What is the final temperature?

③ 13. Ten pounds of hydrogen expand at a constant pressure of 75 lbs. per sq. in. while 1500 B.t.u. are added to it. The initial temperature was 60° F. Compute final temperature, volume, and foot-pounds of external work done.

14. Compute the point of intersection of the curve of Problems 4b and 4c.

15 (a). Compute the foot-pounds of work done under the curves of Problems 2 and 3 between $V_1 = 50$; $V_2 = 200$ cu. ft. and the V -axis. (b) Compute the B.t.u. added to the gas in each case.

16. Ten pounds of hydrogen received 500 B.t.u. while it expanded along the polytropic $PV^{1.3} = K$; $p_1 = 200$ lbs. per sq. in., $t_1 = 125^{\circ}$ F. (a) Find p_2 , V_2 , and t_2 . (b) Compute the foot-pounds of external work done between the curve and the V -axis. (c) Does the B.t.u. content of the gas increase or decrease and how much?

17. A polytropic curve passes through the two points whose coördinates are given below. Compute the value of n .

$$p_1 = 14.7 \text{ lbs. per sq. in.}, \quad V_1 = 120 \text{ cu. ft.},$$

$$p_2 = 42.0 \text{ lbs. per sq. in.}, \quad V_2 = 50 \text{ cu. ft.}$$

18. Find the difference in the foot-pounds of work done during adiabatic and isothermal expansion for 5 lbs. of air from $p_1 = 125$ lbs. per sq. in. to $V_2 = 100$ cu. ft. $t_1 = 100^{\circ}$ F. for both expansions. What is the final temperature and pressure in each case?

19. Compute p_2 ; V_2 ; B.t.u. supplied; change in internal energy in B.t.u.; and external work in foot-pounds for each case when 5 lbs. of air at $p_1 = 30$ lbs. per sq. in. is heated from $t_1 = 70^{\circ}$ F. to $t_2 = 750^{\circ}$ F. (a) When the temperature rise occurs at constant pressure. (b) When the temperature rise occurs at constant volume.

20. Compute the values of c_p and c_v for helium using the following constants: $R = 386.3$; $k = 1.667$.

21. The density of CO_2 at a pressure of 14.696 lbs. per sq. in. and 32° F. is 0.1227 lbs. per cu. ft. Compute the value of R .

22. Discuss the equation of state and the equation of the path of a gas as to similarities, dissimilarities, and limitations of each.

CHAPTER IX

AIR COMPRESSION

66. Introduction. Compressed air has been used on a commercial scale in industry since about 1860. The first important compressed-air plant was set up to aid in driving the Mont Cenis tunnel through the Alps. At the present time the most extensive use of compressed air is made in tunneling and underground mining because the explosion danger by its use is practically nil, and it is an appreciable aid to ventilation. Before the days of the electric motor, compressed air furnished about the only method of transmitting power long distances to small motors. In coal mines, where there are pockets of explosive gases and accumulations of explosive dust, many serious explosions have been caused by sparks from the electric motors and wiring. Hence in mines where this danger is great, compressed-air drilling motors and locomotives are still used. Compressed air has many other industrial applications, for operating small hoists, small portable motors, and a long list of pneumatic tools, for pumping corrosive liquids, and for pumping water and oil wells where the sand is troublesome. Many large compressor installations are used in compressing natural and artificial gas for shipping long distances by pipe line. Considerable quantities of a very light gasoline are also obtained from natural gas in certain fields by the compression process requiring large compressors.

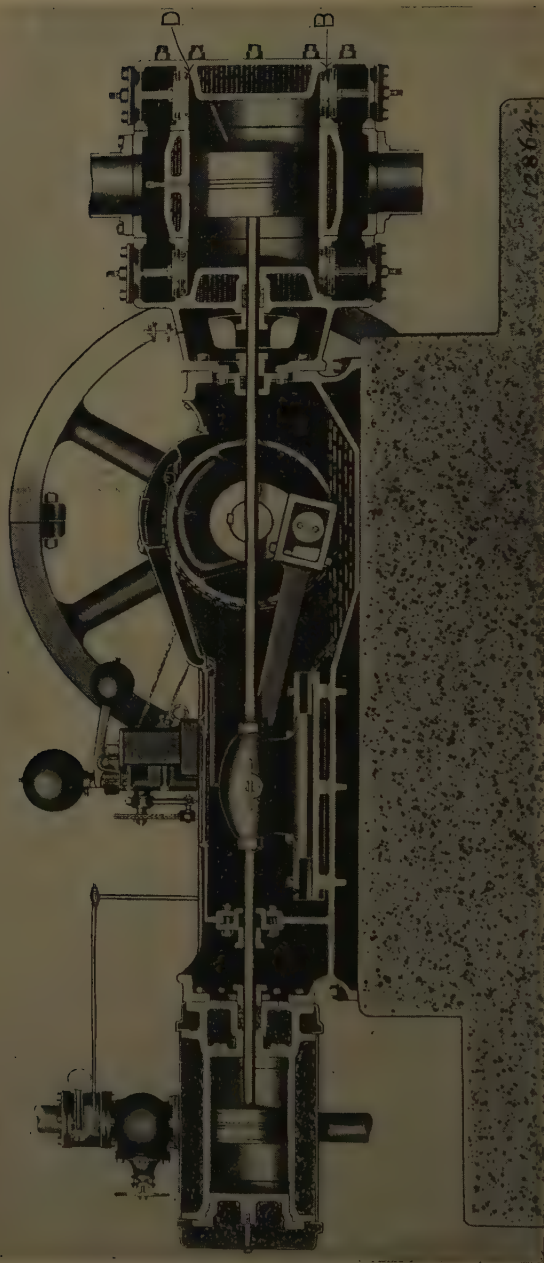
67. The Air Compressor. In Figure 13 is shown a longitudinal, sectional view of a steam-engine-driven displacement air compressor. The essential features of the air end are: a water-jacketed cylinder, piston, spring-loaded intake and discharge valves, and intake and discharge-pipe flanges, and piston-rod.

68. Definitions. There are several terms much used in reference to engines and compressors which should be defined. Any mechanism in which a cylinder containing a movable piston constitutes the main functioning elements is called a *displacement*

machine. One complete movement of the piston from its extreme position at one end of the cylinder to the other end is called a **stroke**. Two full strokes are performed during one complete revolution of the crankshaft. The end of the cylinder adjacent to the crosshead and crank is called the **crank end**. The opposite end is called the **head end**. Any cylinder in which the cycle of changing volume and pressure is carried out on both sides of the piston is a **double-acting machine**. In a **single-acting machine** the cycle is carried out in only one end of the cylinder. The engine and compressor shown in Fig. 13 are both double-acting. It is desirable, in all except the smaller machines, to make them double-acting because the maximum capacity of a double-acting machine is nearly double that of a single-acting machine per unit weight; consequently space requirements, first cost, and operation costs are correspondingly lower.

Obviously the piston, in its sweep through the cylinder, cannot be permitted to bump solidly against the cylinder heads at the ends of the stroke. Also, the air passages leading into and away from the cylinder add a small volume through which the piston is unable to pass. All this unused volume at the ends of the cylinder and in the ports is called the **clearance volume**, or simply the **clearance**. The quantity of clearance for each end of the cylinder is generally stated as a percentage of the volume displaced by the piston in one stroke. Clearance is desirable also because it furnishes a space in which to confine a small volume of air which may act as a cushion in bringing the heavy reciprocating parts to rest without shock at the ends of the stroke. But clearance is a necessary evil because it reduces the useful effect of the piston. The cylinder has to be made larger and heavier, and consequently more costly to allow for the clearance loss. This is especially true in the case of air compressors. In designing and building air compressors, every effort is made to reduce the clearance to the smallest possible value. In the best machines it varies from 2 to 5 per cent of the volume displaced by one stroke of the piston.

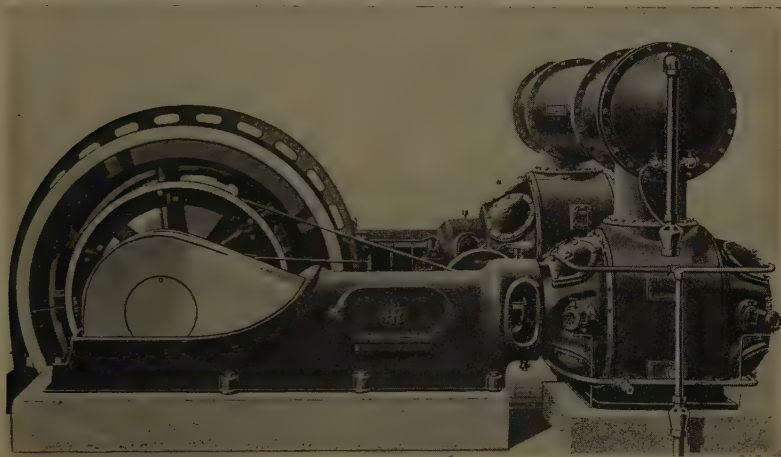
When the entire pressure rise is developed in one cylinder, the compressor is **single-stage**. When the pressure increase occurs



(Ingersoll-Rand Co.)

FIG. 13.—LONGITUDINAL SECTION THROUGH STEAM-DRIVEN AIR COMPRESSOR

partly in each of two cylinders in series, it is called a *two-stage compressor*. Figures 14 and 15 show the arrangement of a two-stage machine. Two single-stage cylinders may be placed side by side and driven by the same crankshaft with the cranks set either 90 degrees or 180 degrees apart. Such an arrangement is called a *duplex compressor*. Compressors are usually made single-stage, single or duplex, for air pressures below 150 pounds per square inch. They are generally made with two stages for air pressures between 150 and 500 pounds per square inch; and three- or four-stage machines for higher pressures.



(Ingersoll-Rand Co.)

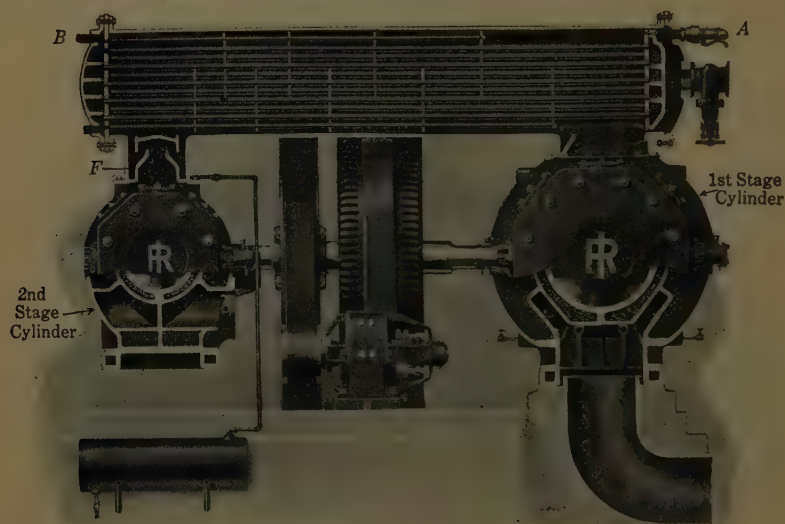
FIG. 14.—TWO-STAGE MOTOR-DRIVEN AIR COMPRESSOR

Free air is air measured at the pressure and temperature of the atmosphere at the place of measurement. *Standard air* is air measured at some specific pressure and temperature. The standard generally adopted by engineers is air measured at a pressure of 30.00 inches of mercury and at a temperature of 60 degrees Fahrenheit. The height of the mercury in the barometer should be 30.00 inches when the temperature of the mercury is 32 degrees Fahrenheit or reduced to that basis by computations. The unit of volume is generally the *cubic foot*.

Air-compressor capacities to deliver air are generally stated in *cubic feet of piston displacement per minute*. The actual volume

of air delivered is always less than the piston displacement. Usually it is between 80 per cent and 90 per cent of the displacement.

69. Water Jackets and Intercoolers. All air-compressor cylinders are surrounded by a jacket in which cooling water is circulated to absorb and carry away most of the heat produced by the work of compression. The jackets are provided about the cylinder barrel and in the cylinder heads as shown in Figs. 13, and 15.



(Ingersoll-Rand Co.)

FIG. 15.—SECTIONS THROUGH INTERCOOLER AND CYLINDERS

In multistage compression the air is cooled still more by being passed through an intercooler placed between the stages. The intercooler is a heat exchanger consisting of a cast-iron shell in which is placed a nest of small copper or brass water tubes. Air from the first-stage cylinder enters the intercooler shell at the right of Fig. 15. The air flows over and around the outside of the nest of tubes. The large volume of the shell aids in dampening out the pulsations in the air stream before it enters the second-stage cylinder. The air flows in a curved path around a series of baffles, crossing and recrossing the tubes several times. Cold water flows through the tubes absorbing the heat from the air. The spacing between the baffles is reduced as the air progresses to the left. This compensates for the decreased

volume of the air as it is cooled and maintains a high, uniform velocity of air flow. One of the important conditions necessary to obtain rapid and efficient heat transfer from a gas through a metal surface to water is that both the gas and the water must flow at suitably high velocities.

The cold, circulating water enters the left-hand end of the intercooler near the bottom, passes to the right through the bottom row of tubes and back to the left through the next higher row of tubes. It finally leaves through the discharge pipe *B* at a temperature 10 to 50 degrees higher than the entering temperature. A baffle plate separator *F* is placed in the air stream to remove the water condensed from the air. This water is present in atmospheric air by virtue of its humidity. It is caught in the tank located below the second-stage cylinder, which tank is emptied from time to time. The intercooler is an important aid in reducing the necessary work of multistage air compression.

70. Compressor Action. The events taking place within a compressor cylinder may best be followed by reference to Figs. 13 and 16. Figure 16 is a conventionalized area on the *PV*-plane representing the cycle of events for one end of the compressor cylinder. As the piston is driven to the right, the spring-loaded intake valve *B* opens automatically. Air flows into the cylinder from the atmosphere to fill the cylinder volume behind the piston as it moves along. The high air pressure above the discharge valve *D* holds it closed during this event. The pressure and volume history in the cylinder during the air-intake or suction stroke for the head end is represented in Fig. 16 by the horizontal line *HA*. This shows that the pressure within the cylinder remains constant at a value slightly below that of the atmosphere. But the volume of air in the cylinder increases. At the end of the suction stroke, the motion of the piston is immediately reversed. The

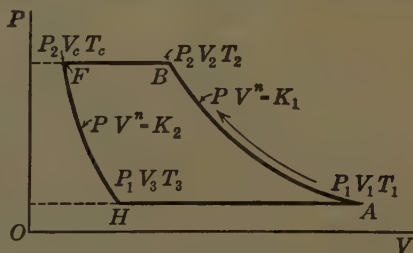


FIG. 16.—CONVENTIONALIZED *PV*-
DIAGRAM

intake valve closes, due to the action of the spring, trapping the air in the cylinder. The returning piston reduces the volume with a consequent rise in pressure along the curve AB . If the water jacket of the cylinder were made of perfect heat-conducting walls, it would be able to remove all the heat due to the work of compression just as rapidly as it was generated. Hence the temperature of the air during the compression period AB would remain constant and the compression curve would be an isothermal. If, on the other hand, the cylinder and piston materials were perfect non-conductors of heat, the compression curve would be an adiabatic.

In practice, neither of these ideal conditions can be realized; therefore the compression curve is not isothermal, nor is it

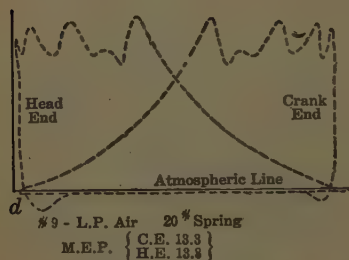


FIG. 17.—FIRST-STAGE CARDS

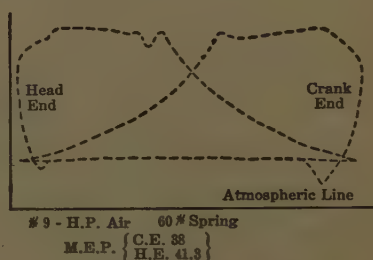


FIG. 18.—SECOND-STAGE CARDS

adiabatic. It is a curve which lies between them. It is a polytropic curve for which the value of the exponent n generally falls between 1.2 and 1.35. At some point B the pressure in the cylinder has risen to a value slightly greater than that in the discharge pipe. The discharge valve D is then forced open against its spring, and the compressed air is forced into the discharge pipe as the piston completes its return stroke. The pressure-volume history during the delivery period is represented by the line BF . When the piston starts on the next suction stroke, the pressure in the clearance space quickly falls, the discharge valve is forced down by the combined action of its spring and the high-pressure air above it. The small volume of air remaining in the clearance space reexpands along the curve FH of Fig. 16. At the point H , the pressure in the cylinder has

fallen slightly below atmospheric as shown more clearly at *d*, Fig. 17. The suction-valve *B* again opens and a new cycle of events begins. The same cycle of events has been going on in the opposite end of the cylinder simultaneously, but out of phase with it by 180 degrees of crank angle. The complete pressure-volume history for the two ends of the cylinder is represented by two *PV*-diagrams superimposed as shown in Figs. 17 and 18 as taken from an actual air compressor. These histories are recorded by means of an indicator which is described on page 310. The wavy discharge and suction lines are due to the

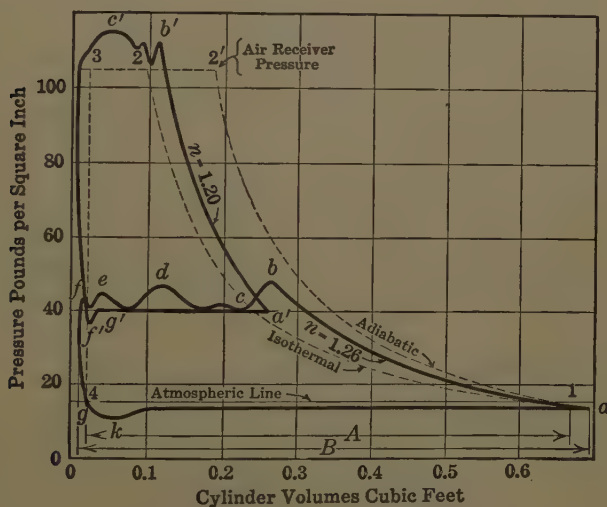


FIG. 19.—COMBINED *PV*-DIAGRAM

effects of the springs and inertia of the intake and discharge valves.

Figure 19 is a combined diagram showing, to scale, the correct values of pressure, volume, and work for the first- and second-stage cylinders of a two-stage compressor the actual cards for which are shown in Figs. 17 and 18. A study of the various areas of the diagrams reveals much useful information. The area *abcdefgk* represents the work done in one end of the cylinder while compressing-and-delivering a cylinder full of atmospheric air in the first stage. The area *a'b'c'd'f'g'* represents the work

done per cycle in compressing-and-delivering air in one end of the second-stage cylinder. If the compression had taken place entirely in one cylinder along an *adiabatic* curve, the total work done to compress and deliver the same weight of air would be represented by the area 12'34. If the compression had been *isothermal* and all in one cylinder, the work would be represented by the area 1234. An inspection of the various areas shows that the isothermal program of compression requires the expenditure of the smallest amount of work. Hence that is the ideal for which to strive. Isothermal compression in a cylinder with a piston made of a perfect heat-conducting material and with complete water cooling, without clearance, friction, or inertia of valves, and with instantaneous valve action, is the ideal of perfection to be desired in an air compressor. The *PV*-diagram from such an ideal compressor is shown by area *abcd* and in Fig. 20. Such ideal performance cannot be realized in practice but it may be approached more or less closely. Dividing the compression up into stages with intercoolers between them, and providing water jackets for the cylinders, are the best expedients so far discovered for approaching the ideal cycle of compression.

71. Work of Compressing-and-Delivering Air. Case I. The compression-and-delivery of air will first be considered for the

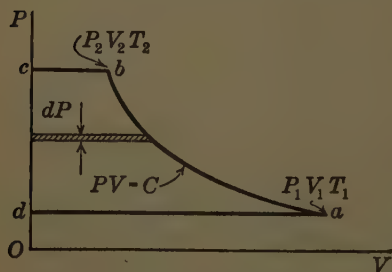


FIG. 20.—IDEAL COMPRESSOR
DIAGRAM

ideal conditions of isothermal compression in a compressor without clearance and having a frictionless piston (see Fig. 20). The area *abcd* represents the mechanical work required to compress-and-deliver a cylinder full of air from $P_1V_1T_1$ to $P_2V_2T_2$. By authority of equation (68), page 57, the work of the ideal compressor

cycle is

$$(160) \quad W_i \int_{P_1}^{P_2} V dP = P_1 V_1 \log_e \frac{P_2}{P_1}.$$

For isothermal compression let

$$\frac{P_2}{P_1} = \frac{V_1}{V_2} = r,$$

then

$$(161) \quad W_i = P_1 V_1 \log_e r.$$

The quantity, r , is known as the *ratio of compression*. It is always a number greater than unity. It corresponds to the ratio of expansion, r , discussed on page 80 above. Notice that equation (161) is exactly similar to equation (134).

Case 2. In the case of the actual compressor, the compression curve is not an isothermal, but it is a polytropic for which the value of n varies between 1.2 and 1.35 (see Fig. 21). Hence the work of *compressing-and-delivering* a cylinder full of air when there is no clearance is represented by area $abcd$, and is given by the formula

$$(162) \quad W' = \int_{P_1}^{P_2} V dP.$$

Since the equation for this compression curve is

$$PV^n = K,$$

one has

$$(163) \quad V = \frac{K^{1/n}}{P}.$$

Substituting (163) in (162), one obtains

$$(164) \quad W' = K \int_{P_1}^{P_2} \frac{dP}{P^{1/n}} = \frac{n}{n-1} (P_2 V_2 - P_1 V_1).$$

It should be observed that the result given by equation (164) does not equal that given by equation (118), p. 77. Equation (27) may be transformed to give

$$(165) \quad P_2 V_2 = P_1 V_1 \frac{T_2}{T_1}.$$

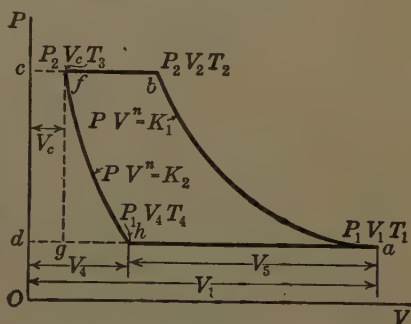


FIG. 21.—CONVENTIONALIZED PV-DIAGRAM

The relations between pressures and temperatures for a polytropic change in Table 12, page 81 give

$$(166) \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{(n-1)/n}.$$

Substituting (165) and (166) in (164) and simplifying, one finds

$$(167) \quad W' = \frac{n}{n-1} \left[P_1 V_1 \left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right].$$

If, for polytropic compression

$$(168) \quad \frac{P_2}{P_1} = y,$$

equation (167) becomes

$$(169) \quad W' = \frac{n}{n-1} P_1 V_1 (y^{(n-1)/n} - 1).$$

Note that the *ratio of pressures for polytropic compression*, y , is always a number greater than unity. Also note that when the exponent for the compression curve does not equal unity,

$$(170) \quad \frac{P_2}{P_1} \text{ does not equal } \frac{V_1}{V_2},$$

but

$$(171) \quad \frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^n.$$

Equations (167) and (169) give the number of foot-pounds of work which must be expended in compressing V_1 cubic feet of air from a pressure of P_1 pounds per square foot to a pressure of P_2 pounds per square foot and delivering it against the higher pressure in a compressor without clearance.

Case 3. An actual compressor can never be built with zero clearance. The effect of clearance is to reduce the volume of free air delivered per stroke from ad to ah (Fig. 21) due to the re-expansion of the air caught in the clearance along the curve fh . Then the work required to compress-and-deliver a volume of air equal to ah is equal to the area $abcd$ minus the area $hfcd$. If one assumes that the value of the exponent is the same for the com-

pression and expansion curves, then for the actual compressor with clearance, the work required is

$$(172) \quad W_a = \left[P_1 V_1 \frac{n}{n-1} (y^{(n-1)/n} - 1) \right] - \left[P_1 V_4 \frac{n}{n-1} (y^{(n-1)/n} - 1) \right].$$

But

$$(173) \quad V_5 = V_1 - V_4.$$

Hence equation (172) may be reduced to the form

$$(174) \quad W_a = P_1 V_5 \frac{n}{n-1} (y^{(n-1)/n} - 1).$$

In equation (174), W_a represents the work in foot-pounds required to **compress-and-deliver** V_5 cubic feet of free air from a pressure of P_1 pounds per square foot to P_2 pounds per square foot. If V_5 cubic feet of free air were compressed-and-delivered per minute, the net air horsepower required would be

$$(175) \quad \text{A. hp.} = \frac{W_a}{33\,000}.$$

72. Volumetric Efficiency. If the value of V_5 in Fig. 21 is the same as the value of V_1 in Fig. 21, the volume of air compressed-and-delivered is the same; and the work required to compress-and-deliver it is the same provided n , P_1 , and P_2 are the same in both cases. Hence the work required to compress-and-deliver a given quantity of air is independent of the clearance. But to deliver a given volume of air per stroke, a compressor with clearance must be larger than one without clearance. Just how much larger may be determined as follows. In Figure 21, the length of the line cf represents the clearance volume V_c , and the length of the line ag represents the volume displaced by the piston in one stroke. Then let

$$(176) \quad c = \frac{\text{Vol } cf}{\text{Vol } ag}, \quad (\text{a decimal}).$$

When the piston displacement ag is taken equal to 1.00, c is a decimal number representing the clearance volume. Then

$$(177) \quad \text{volume } dh = V_4 = cy^{1/n}.$$

Now remembering that the volume ag displaced by the piston is considered to be unity, one sees that the volume

$$(178) \quad gh = dh - c,$$

and

$$(179) \quad gh = c(y^{1/n} - 1).$$

The volume of free air actually delivered per stroke equals the volume ah = volume ag - volume gh (Fig. 21). But since the piston displacement ag is considered to be unity,

$$(180) \quad \text{volume } ah = 1.00 - c(y^{1/n} - 1).$$

The *volumetric efficiency* of an air compressor may be defined as follows:

$$(181) \quad e_v = \frac{\text{volume of free air delivered per stroke}}{\text{volume of piston displacement per stroke}}.$$

Therefore, when the volume of the piston displacement is considered to be unity,

$$(182) \quad e_v = 1.00 - c(y^{1/n} - 1).$$

In Fig. 19, the ratio of volume A to volume B , as indicated by the dimension lines, gives the *apparent volumetric efficiency* from the compressor diagrams. The value of efficiency obtained by solving equation (182) is the *theoretical volumetric efficiency*. Since there is always more or less leakage of air past the piston, valves, and packings of an actual air compressor; the actual volumetric efficiency determined by careful test is always less than either the apparent or the theoretical values. The actual value is also smaller because the temperature of the air in the cylinder at the end of the suction stroke is higher than the temperature of the outside atmosphere. Values of volumetric efficiency vary from 50 per cent to 75 per cent in small, cheap, single-stage machines. In large, well-designed, well-built machines, it varies from 80 to 93 per cent.

73. Efficiency of Compression. The ratio of the work required to compress isothermally and deliver air by an ideal compressor without clearance to the work required by an actual compressor to compress and deliver the same volume of free air is called the *efficiency of compression*.

Efficiency of compression depends upon the discharge pressure, upon the amount of air leakage, and upon the grade of workmanship employed in constructing the compressor. Small cheaply-built machines, for pressures up to 100 pounds, will show upon test, efficiencies of compression ranging between 50 and 70 per cent. In large high-class machines, the efficiency of compression is often as high as 90 per cent.

74. Intercooler Pressure. The most desirable intercooler pressure at which to operate a two-stage compressor may be determined, approximately, as follows. The result desired is an intercooler pressure which will require the smallest expenditure of work in compressing-and-delivering a given quantity of free air. The intercooler pressure in a two-stage machine is the delivery pressure for the first stage, and it is the intake pressure for the second-stage cylinder. The most desirable intercooler pressure may be readily computed upon the following assumptions: first, that the intercooling is complete, so that the air temperature is reduced to that at which it entered the first-stage cylinder; second, that the value of the exponent n is the same for both stages; third, that there are no leakage losses. The following symbols will be required.

P_1 = first-stage intake pressure, lbs. per sq. ft.,

P_2 = first-stage discharge pressure; it is also the first-intercooler pressure and the second-stage intake pressure, lbs. per sq. ft.,

P_3 = second-stage discharge pressure; it is also the second-intercooler pressure and the third-stage intake pressure, lbs. per sq. ft.,

P_4 = third-stage discharge pressure, lbs. per sq. ft.,

V_1 = volume of air at P_1 taken into the first-stage cylinder per stroke, cu. ft.

V_2 = volume of air at P_2 , in cu. ft.,

V_3 = volume of air at P_3 , in cu. ft.,

w_1 = foot-pounds of work done per stroke in the first-stage cylinder,

w_2 = foot-pounds of work per stroke in the second-stage cylinder,

$W = w_1 + w_2$,

$$x = \frac{n-1}{n}.$$

Then, by equation (169), the following relations are obtained:

$$(183) \quad w_1 = \frac{P_1 V_1}{x} \left[\left(\frac{P_2}{P_1} \right)^x - 1 \right],$$

$$(184) \quad w_2 = \frac{P_2 V_2}{x} \left[\left(\frac{P_3}{P_2} \right)^x - 1 \right],$$

$$(185) \quad W = \frac{P_1 V_1}{x} \left[\left(\frac{P_2}{P_1} \right)^x + \left(\frac{P_3}{P_2} \right)^x - 2 \right].$$

Equation (185) is correct since for isothermal change $P_1 V_1 = P_2 V_2$.

Pressures P_1 and P_3 are fixed by the conditions of the problem but the value of P_2 may be varied at will. The work W will be a minimum when

$$(186) \quad \left(\frac{P_2}{P_1} \right)^x + \left(\frac{P_3}{P_2} \right)^x$$

has a minimum value. Differentiating with respect to P_2 , equating to zero, solving, and simplifying the result, one finds that the minimum work is

$$(187) \quad P_2 = \sqrt{P_1 P_3},$$

or

$$(188) \quad \frac{P_2}{P_1} = \frac{P_3}{P_2} = \left(\frac{P_3}{P_1} \right)^{.5}.$$

By combining (185) and (188), one obtains the simpler result

$$(189) \quad W = \frac{2P_1 V_1}{x} \left[\left(\frac{P_3}{P_1} \right)^{.5x} - 1 \right].$$

A similar series of steps may be carried out for a three-stage compressor, and it will be found that the minimum work of compression-and-delivery is

$$(190) \quad W = \frac{3P_1 V_1}{x} \left[\left(\frac{P_4}{P_1} \right)^{x/3} - 1 \right],$$

and for m stages

$$(191) \quad W = \frac{mP_1 V_1}{x} \left[\left(\frac{P_{m+1}}{P_1} \right)^{x/m} - 1 \right].$$

75. Illustrative Compressor Problem. The following data applies to a certain single-stage double-acting compressor when operated at 125 r.p.m. (revolutions per minute): cylinder diameter 18 ins.; length

of piston stroke 16 ins.; clearance volume is 8 per cent of piston displacement; the value of the exponent n is 1.25; p_1 equals 14.7 lbs. per sq. in.; p_2 equals 80 lbs. per sq. in. The net air work, the net air horsepower, and the volumetric efficiency are found as follows (refer to Fig. 21):

$$y = \frac{P_2}{P_1} = \frac{80}{14.7} = 5.44,$$

$$V_1 \text{ (including clearance)} = \frac{1.08 \times 16 \times 9^2 \times 3.1416}{1728} = 2.545 \text{ cu. ft.},$$

$$\frac{n-1}{n} = x = \frac{0.25}{1.25} = 0.20,$$

$$\frac{n}{n-1} = \frac{1}{x} = \frac{1}{0.20} = 5.00,$$

$$\frac{1}{n} = \frac{1.00}{1.25} = 0.80,$$

$$V_4 = V_1 y^{1/n} = 0.1885(5.44)^{.8} = 0.731 \text{ cu. ft.},$$

$$y^{n-1/n} - 1 = (5.44)^{.2} - 1 = 0.403,$$

$$V_5 = 2.545 - 0.731 = 1.814 \text{ cu. ft.},$$

$$W = 144 \times 14.7 \times 1.814 \times 5 \times 0.403 = 7737 \text{ ft. lbs. per stroke.},$$

$$\text{net air horsepower} = \frac{2 \times 125 \times 7737}{33\,000} = 58.6,$$

and the volumetric efficiency is

$$e_v = 1.00 - 0.08(5.44^{.8} - 1) = 0.77 = 77\%.$$

76. Dimensions of an Air-Compressor Cylinder. When an air compressor is selected for a given service, the cubic feet of free air per minute to be delivered, the intake, and the delivery pressures are generally known. The size and type of compressor determine approximately the amount of clearance volume and the approximate number of revolutions per minute. Air-compressor speeds vary from 100 r.p.m. to 500 r.p.m., depending upon the size of the compressor and type of valves. Piston speeds much used in design of air compressors range from 350 to 700 ft. per min. (Piston speed is the distance travelled in ft. per min. by a point on the piston. It equals the product of stroke in feet, the number of revolutions per minute, and the factor two.) The higher r.p.m. and lower piston-speeds are confined

to the smaller sizes. The dimensions of the compressor cylinder may be determined as illustrated in the problem worked out below.

It is desired to compress-and-deliver 1000 cu. ft. of free air (at 75° F. and 14.5 lbs. per sq. in.) per minute. The delivery pressure is to be 110 lbs. per sq. in. abs. A single-stage double-acting machine having about 4 per cent clearance and operated at 250 r.p.m. would be suitable. Assuming that n equals 1.30, the volumetric efficiency of the cylinder is

$$e_v = 1.00 - c(y^{1/n} - 1) = 1.00 - .04 \left[\left(\frac{110}{14.5} \right)^{1/1.3} - 1 \right] = 0.850$$

Therefore the piston displacement will have to be

$$\frac{1000}{0.85} = 1176 \text{ cu. ft. per min.}$$

Let N = revolutions per minute,

d = piston diameter in inches,

l = length of piston stroke in inches.

Then piston speed equals

$$(192) \quad 2lN = 600 \times 12 = 7200 \text{ ins.}$$

and

$$(193) \quad \frac{2\pi d^2 l N}{4 \times 1728} = 1176.$$

Substituting (192) in (193) the result is

$$(194) \quad d = 18.95 \text{ ins.} = 19.0 \text{ ins., approximately.}$$

This leaves the length of stroke and r.p.m. undetermined. The only limitation as indicated by (192) is that their product equals 3600. But by long experience in building air compressors, it has been found that the best length of stroke is from $0.7d$ to $1.5d$, and the best speed range is from about 125 r.p.m. for the large units to 350 or 400 r.p.m. for the small units. Choosing a middle-of-the-road value of l as 20 ins. and solving (192) for N , we find the result

$$(195) \quad N = \frac{3600}{l} = \frac{3600}{20} = 180 \text{ r.p.m.}$$

This gives a compressor in line with modern steam-engine-driven units for this capacity. A 15-inch stroke and a corresponding speed of 240 r.p.m. would be suitable if the compressor were driven by an electric motor.

PROBLEMS

1. Compute the items called for below from the given data for a single-stage double-acting air compressor. The cylinder dimensions are 20 in. diameter by 18 in. stroke; r.p.m. 225; clearance = 3 per cent; the suction pressure, $p_1 = 14.4$ lbs. per sq. in.; the delivery pressure, $p_2 = 115$ lbs. per sq. in.; the exponent for the actual compression curve = 1.35. Compute the volumetric efficiency, the actual air horsepower, and the ideal horsepower for isothermal compression.

2. It is desired to compress-and-deliver, in two stages, 2000 cu. ft. of free air per minute from $p_1 = 14.5$ lbs. per sq. in. and $t_1 = 75^\circ$ F. to a pressure $p_3 = 300$ lbs. per sq. in. Determine the best intercooler pressure and the net air horsepower if $n = 1.25$ for both stages.

3. Compute the delivery capacity in cubic feet of air per minute at sea-level conditions for different altitudes as indicated below; if at sea level ($p_1 = 14.7$ lbs. per sq. in. and $t_1 = 60^\circ$ F.) a compressor delivers 1000 cu. ft. of free-air per minute. Assume the initial air temperature to be 60° F. at all elevations.

- a) 2000 ft. elevation $p_1 =$ approx. 13.6 lbs. per sq. in.,
- b) 5000 ft. elevation $p_1 =$ approx. 12.2 lbs. per sq. in.,
- c) 8000 ft. elevation $p_1 =$ approx. 10.6 lbs. per sq. in.

4. Compute and arrange in convenient tabular form volumetric efficiencies for the following conditions (each value of c to be used with every value of n ; 16 answers).

c	n	η
0.01	1.00	7.0
0.03	1.20	7.0
0.05	1.30	7.0
0.07	1.35	7.0

5. Supply all the missing steps between equations (185) and (189).

6. Determine suitable cylinder dimensions for a single-stage double-acting air compressor which will deliver 1500 cu. ft. of free air per minute under the following conditions: $p_1 = 14.25$ lbs. per sq. in.; $p_2 = 90$ lbs. per sq. in.; $t_1 = 80^\circ$ F.; clearance = 5 per cent.

7. Determine the most economical intercooler pressure for a two-stage compressor when $p_1 = 14.5$ lbs. per sq. in. and $p_3 = 225$ lbs. per sq. in.

8. Supply all the necessary steps in the development of equation (190).

CHAPTER X

VAPORS AND THEIR PROPERTIES

77. Vapors. A gas whose temperature is relatively near the liquefaction temperature is called a vapor. A vapor is also considered to be a gas which may be liquefied by an increase in pressure alone. A reduction in temperature, as well as an increase in pressure, is necessary in order to liquefy a true gas. The cardinal properties of a vapor are six in number instead of five as for gases. They are: pressure, volume, temperature, entropy, internal energy, and quality. The equation of state for vapors and the equations between all the cardinal properties are much more complex than for perfect gases or even for the true gases. The values of all the other cardinal properties of vapors, except quality, may be expressed as functions of the temperature only.

If the value of R for water vapor in the equation of state for a perfect gas be computed from the molecular gas constant and the molecular weight of water, it is found that

$$(196) \qquad R = \frac{1545}{18} = 85.83$$

In Table 13 are given values of R in equation (32), page 44, computed from values of P , v , and T prepared by J. H. Keenan ¹ in a report upon the development of new steam tables and charts. These values represent the latest and most authentic information available upon the properties of water vapor. An examination of the table shows that the values of R for water vapor computed from laboratory observations approach at low pressure and high temperature the theoretical value for a perfect gas; but it never seems to quite reach that value. Also the value of R for vapors does not have anything like a constant value.

Exact equations, which among others have been proposed in

¹ See Steam Tables and Mollier Diagram, pub. by the A. S. M. E., 1930.

TABLE 13
VALUES OF R FOR WATER VAPOR

PRESSURE LBS. PER SQ. IN. ABS.	CONDITION OF THE STEAM				
	DRY AND SATURATED	SUPERHEAT DEGREES F.			
		100	200	300	400
15	84.6	84.6	85.0	85.2	85.3
100	81.0	83.0	84.0	84.5	84.9
200	78.2	81.4	83.0	83.3	84.1
500	72.0	72.3	80.2	81.7	82.8
1200	60.4	70.5	75.0	72.8	78.7
3200	23.8	58.1	66.1	70.0	—

the past to represent the relations between some of the cardinal properties for water vapor, are reproduced below.¹

$$(197) \quad p(v + 0.088) = 85.87 T - p(1 + 0.0006 p) \times \frac{47.795 \times 10^{12}}{T^5},$$

$$(198) \quad E_i = T(0.2566 + 5 \times 10^{-5} T) - (p + 24 \times 10^{-5} p^2) \times \frac{44.252 \times 10^{12}}{T^5} + 887.3,$$

and

$$(199) \quad N = 0.8451 \log T + T \times 10^{-4} - 0.2542 \log p - (p + 3 \times 10^{-4} p^2) \frac{44.252 \times 10^{12}}{T^5} - 0.3960.$$

In the above equations

p = pressure in lbs. per sq. in. abs.,

v = volume of one pound of steam in cu. ft.,

T = absolute temperature, degrees Fahrenheit,

E_i = internal energy in B.t.u.,

N = total entropy for one pound of steam.

Equations of the form of (197) are partly rational and partly empirical. That is, they are built up to fit curves plotted from values observed in laboratory investigations. Another form of equation, entirely empirical, is frequently employed. It is

$$(200) \quad p = A \pm BT \pm CT^2 \pm DT^3 + \dots$$

¹ See G. A. Goodenough, *A discussion of certain thermal properties of steam*, Transactions, A. S. M. E., vol. 34 (1912), pp. 507-560.

In this form the values of the constants are selected to fit the curves obtained from experimental observations. Many forms of equations and many different values of the constants have been proposed by investigators for the properties of steam and other vapors since the time of Henry V. Regnault,¹ who was one of the first to carry out careful and authentic studies upon the values of the properties of vapors. The diversity in the results are adequately explained by the fact that it is extremely difficult to obtain accurate and correct measurements of the quantities involved. Even if the correct values were available, the equations in which they would be employed are too cumbersome for frequent and rapid solution by engineers. This problem is met satisfactorily by the preparation and use of elaborate vapor tables which are arranged in convenient form.²

78. Formation of Vapors. Vapors are considered to exist in three different stages; *wet vapor*, *dry and saturated vapor*, and *superheated vapor*. Perhaps the process of vaporization of a liquid may be best explained by considering, step by step, what happens when a pound of water is evaporated into steam at constant pressure, which is the condition in a steam boiler. Imagine the pound of water at 32° F. to be confined in a cylinder under a moveable, weighted piston as at *a*, Fig. 22. The piston with its weight and the pressure of the atmosphere produce a total pressure of *p* pounds per square inch upon the liquid and vapor. Heat is added to the water, its temperature gradually rises to a value called the *boiling point*. The heat energy absorbed during this step is called *heat of the liquid*, or sometimes sensible heat, because it affects the thermometer or is noticeable to the sense of touch. The boiling-point temperature varies in accordance with the pressure to which the liquid is subjected.

¹ See Memoirs de l'Institut de France, vol. 21, 1847.

² J. H. Keenan, Steam Tables and Mollier Diagram.

See G. A. Goodenough, *Properties of Steam and Ammonia*.

Tables of Thermodynamics Properties of Ammonia, Circular of the U. S. Bureau of Standards No. 142, 1923.

L. A. Sheldon, *Properties of mercury vapor*, Transactions, A. S. M. E., Vol. 46, 1924, p. 272.

A. S. M. E., *Steam Table Research*, Mechanical Engineering, Feb., 1925, Feb., 1926, Feb., 1927, Feb., 1928, and Feb., 1929.

As the temperature rises from 32° F. to 39.2° F., the space occupied by the water decreases slightly. As the temperature rises above 39.2° F., the volume increases slightly, forcing the loaded piston upward in the cylinder. Suppose the total pressure upon the water produced by the loaded piston and the atmosphere is 100 pounds per square inch. The boiling point for that case is found to be 327.8° F. If heat addition is continued, the water boils violently. Some of it is changed into the vapor

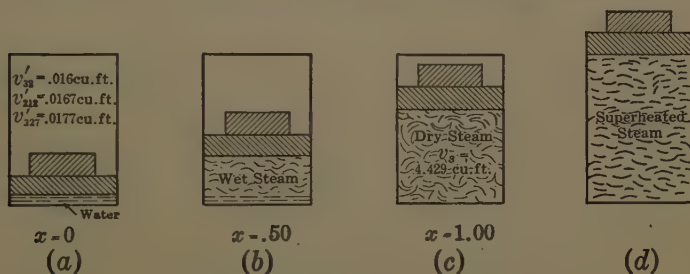


FIG. 22.—VAPORIZATION OF WATER

state and the volume increases very noticeably, forcing the piston upward more rapidly than before the boiling temperature was reached. During vapor formation the temperature remains constant provided the pressure is constant. The heat added during this stage is called *latent heat of evaporation*, or simply *latent heat*. The movement of the piston produces external work since it constitutes a force moving through a distance. Consider the situation when half of the water has been evaporated (Fig. 22b). There is then present in the cylinder one-half pound of H_2O liquid and one-half pound of H_2O vapor. The mixture is called *wet steam*. In that case it is said to be wet steam whose *quality* is 50 per cent. As more heat is added the evaporation continues. As the proportion of vapor in the cylinder increases the quality increases in a direct ratio. When the liquid is 75 per cent evaporated, the quality is 75 per cent. When the evaporation is just complete the quality is 100 per cent, and the pound of steam is then just dry and saturated as represented at Fig. 22c. The volume of one pound of liquid at 32° F. is 0.01602 cubic feet. The volume of one pound of dry saturated vapor at a pressure of

100 pounds per square inch absolute is 4.426 cubic feet. If from the pound of dry and saturated steam whose quality is just 100 per cent the smallest possible quantity of heat energy were extracted, a small part of the steam would be condensed back to the liquid state. If on the other hand the smallest possible amount of heat energy were added, the temperature of the vapor would begin to rise above 327.8° F., the boiling point, and the volume would continue to increase above 4.426 cubic feet. The quality might also be said to have a value greater than 100 per cent. The steam is then called *superheated steam*. (Fig. 22 d.) However, values of quality above 100 per cent are never ascribed to superheated vapors. The amount of superheating is designated by stating the number of degrees Fahrenheit (or Centigrade) the temperature of the steam may be above the boiling point for its pressure. If the superheating be carried to extremely high temperatures, for instance 1000° F. or 2000° F., the steam acquires the characteristics of a true gas and its behavior conforms to the laws for gases. However, at the present time in engineering practice, steam superheated more than 300° F. or 400° F. is seldom used for any purpose. The gas of a complex substance begins to dissociate into its component elements at a temperature at or near 2000° F.; hence there is chemical change as well as a physical one, and the simple laws of thermodynamics are no longer applicable.

TABLE 14¹
BOILING POINTS FOR WATER

PRESSURE, LBS. PER SQ. IN. ABS.	B. P. DEGREES FAHRENHEIT	PRESSURE, LBS. PER SQ. IN. ABS.	B. P. DEGREES FAHRENHEIT
1	101.8	500	467.0
14.7	212.0	800	518.2
100	327.8	1200	567.1
200	381.8	3200	704.9
300	417.3	3226	706.1

From the above discussion the following definitions may be formulated.

¹ See Mechanical Engineering, Feb., 1929, p. 113.

Steam or vapor is a gas at or near its temperature of condensation.

Boiling point of a liquid is the temperature at which the substance begins to change from the liquid to the vapor state. The boiling point is a fixed temperature so long as the pressure of the vapor formed is constant. Boiling points for water vary from 32° F. at a pressure of 0.0885 lbs. per sq. inch absolute to 706° F. at a pressure of 3226 pounds per square inch absolute. (See Table 14.)

Wet steam is a mixture of vapor and its liquid at the temperature of the boiling point.

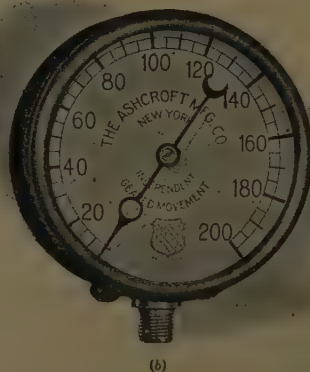
Quality of wet steam or simply quality is a decimal number or a per cent representing the proportion of a mixture of liquid and vapor that is in the vapor state. The value of the quality is considered never to exceed 1.00 or 100 per cent.

Dry and saturated steam is water vapor at the minimum temperature possible for the existing pressure, that is, the boiling point. It may be in contact with or in the presence of its liquid. It is in thermal equilibrium. If the smallest quantity of heat energy were extracted from it, part of the vapor would condense to liquid. If the smallest quantity of heat energy were imparted to it, its temperature would rise above the boiling point, provided no liquid is present. At a given temperature the other cardinal properties of a vapor have only one value. The other cardinal properties of a gas have many values at constant temperature. The specific volume of a dry and saturated vapor is its smallest possible volume at a given pressure and still remain completely in its vapor state. Consequently its density, $1/v$, is the maximum for its pressure. Also the quality is 100 per cent.

Superheated steam is water vapor at a temperature higher than the boiling point of its liquid. Its specific volume is greater and its density less than for dry and saturated steam. At a given pressure or at a given temperature the other cardinal properties may have a series of values. Superheated vapors cannot exist in the presence of their liquids.

The cardinal properties of a vapor are the same as for true gases (see p. 37) and one more, namely, quality.

79. Bourdon Pressure Gage. Steam pressures are ordinarily measured by means of the familiar gage shown in Figs. 23 and 24. Eugene Bourdon, a French maker of aneroid barometers, discovered in 1849 that a curved hollow tube of elliptical cross section changes very slightly its radius of curvature when there is a difference between the pressures inside and outside the tube. When the pressure within the tube is greater than atmospheric, it tends to straighten out. In Fig. 23*a*, *A* is such a tube of ellip-



(The Ashcroft Mfg. Co.)

FIG. 23.—BOURDON PRESSURE GAGE

tical cross-section curved into a circular form. The interior of the fixed end *E* is in communication with the boiler or steam pipe. The closed end *F* is free to move. This movement is transmitted through a linkage *B* to the gage hand or indicator attached to the arbor *C*. The mechanism is then enclosed in a suitable case, Fig. 23*b*, having a graduated scale protected by a glass front through which the indicator may be observed.

The zero of the Bourdon scale is always set at atmospheric pressure. Hence the gage indications are above or below that datum. The gage scale is generally so calibrated that its indications above atmospheric pressure are in pounds per square inch. Pressures below that of the atmosphere, called *vacuums*, are always indicated in inches of mercury. To obtain pressures measured above absolute zero, the barometer reading, reduced to pounds per square inch, must be added to the gage indications.

If

B = the barometer reading in inches of mercury,

g = gage indication in lbs. per sq. in.,

p = pressure in lbs. per sq. in., above absolute zero pressure,

V = vacuum gage indication in inches of mercury below atmospheric pressure,

Then for pressures above atmospheric pressure,

$$(201) \quad p = g + 0.491 B,$$

and for vacuums (pressures below that of the atmosphere),

$$(202) \quad p = 0.491 (B - V).$$

(One cubic inch of mercury at 32° F. weighs 0.491 pounds.)



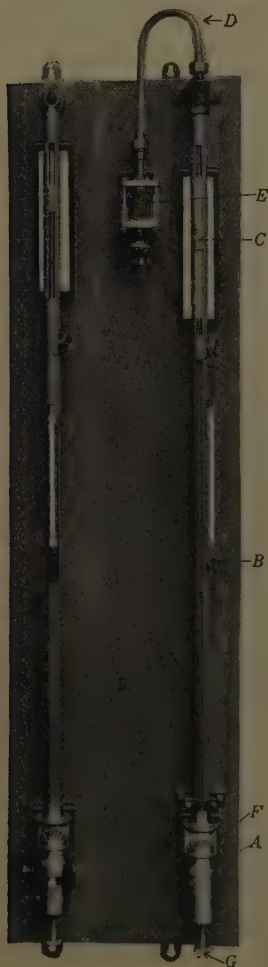
(The Ashcroft Mfg. Co.)

FIG. 24.—COMPOUND BOURDON GAGE

Some gages (Fig. 24) are provided with a scale reading both above and below the zero point. Such a gage is called a *compound gage*. The scale above zero indicates pressures in pounds per square inch gage; the indications below zero are vacuums in inches of mercury.

The Bourdon type of pressure gage gives satisfactory indications between 29 inches of vacuum and 5000 lbs. per sq. in. But its indications are frequently in error from 2 to 5 per cent, making frequent calibration and adjustment necessary. Even a new well-made gage of the Bourdon type, when in the best possible adjustment, seldom has an error less than 1 or 2 per cent.

80. Mercury-Vacuum Gage. A precision mercury gage and mercury barometer, mounted together on a suitable board as shown in Fig. 25, has been developed recently for use in modern



(Precision Thermometer & Instrument Co.)

FIG. 25.—BROMETER. AND PRECISION VACUUM GAGE

steam-turbine plants where precise measurements of vacuum are important. The mercury gage shown at the right in the figure is very similar to an ordinary barometer. The mercury well *A*, mercury column *B*, and the vernier scale for reading the heights of the mercury column are identical to those of the standard barometer which is alongside. However, the top of the mercury-gage column is open and communicates with the condenser through the tube *D*, and the moisture and sediment trap at *E*. A connection on the back of the board leads from the trap directly to the condenser.

The mercury column *B* consists of a glass tube about 32 inches long, open at both ends, and housed in a brass tube which is cut away at the front and back to make the mercury visible. The lower end of the tube dips into the mercury well *A*. The zero of the scale for measuring the height of the mercury column is indicated by the bottom tip of a fixed ivory peg which projects downward in the mercury well, and is visible through the glass window *F*. As the mercury rises in the small glass tube *B*, the surface of the mercury in the well necessarily falls. In order to maintain the

surface at the level of the zero of the scale, the mercury well may be raised by the adjusting screw *G* just as in the case of the standard barometer. The amount of the pressure reduction below atmos-

pheric in the condenser is indicated by the height to which the mercury column rises in the tube *B*. The scale along the tube is usually calibrated in inches and tenths. By means of a vernier at *C*, the nearest thousandths of an inch may be read.

Two precautions should be observed in using mercury columns if correct indications of vacuums or pressures are to be obtained. First, the mercury should be chemically pure and be kept so. Pipes made of copper or its alloys should never be used where there is a possibility of mercury entering them, since mercury will dissolve or amalgamate with them. Second, the mercury tube *B* should always be held in a *vertical position* in order to obtain the full effect of gravity, or if not, the angle of inclination should be measured and proper allowance made for it in determining the pressure from the height of the mercury column.

The instruments shown in Fig. 25 are high grade and correspondingly expensive. For ordinary use, the Bourdon type of vacuum gage or the usual U-tube manometer, which is one of the many inventions of James Watt, will be found satisfactory for most purposes.

81. Vapor Tables. Vapor tables used by American engineers contain values conveniently arranged for 10 or 20 items. There are usually two tables for dry and saturated vapor. The index columns for these two tables are respectively temperature in degrees Fahrenheit and pressure in pounds per square inch absolute. There is then a third table for superheated vapors. The other cardinal properties generally included are volume, internal energy, and entropy. The density, which is the reciprocal of volume, is also included. Then there are certain heat quantities as explained below. English units are generally employed in this country, although the tendency to give parallel values in metric units is growing. The values are all for the weight of one pound of substance. The temperature datum above which all heat quantities are customarily given varies for the different vapors. In ammonia-vapor tables, the datum generally chosen is -40° F. For carbon dioxide, sulphur dioxide, and water (H_2O vapor), the usual datum is 32° F.

The *heat of the liquid* is the heat energy in B.t.u. required to raise the temperature of the liquid from the datum temperature to the boiling point for the pressure under consideration. The quantity is $H + AW$ of equation (72), page 61 for the liquid state. However the AW term is so very small compared to the value of H that it is often neglected.

The *latent heat of evaporation* or simply latent heat is the heat energy in B.t.u. necessary to perform the internal and external work in changing one pound of the liquid at its boiling point into dry and saturated vapor at the same temperature. It is $I + AW$ of equation (72), page 61.

The *total heat* of the vapor or the heat content¹ is the sum of the heat of the liquid and the latent heat. This quantity is called *enthalpy* by some writers. It equals $H + I + AW$ of equation (72) except that the AW term for the liquid is omitted. Equations representing the heat quantities for vapors are readily arranged using the symbols as explained below. *The weight of the substance is always understood to be one pound.* For water vapor, the temperature datum above which heat quantities are measured is always 32° F.

Let

H = total B.t.u. in one pound of dry and saturated steam,

H_w = total B.t.u. of wet steam,

H_s = total B.t.u. of superheated steam,

r = B.t.u. of latent heat of evaporation for one pound,

h = B.t.u. required to raise the temperature of one lb. of liquid from 32° F. to the boiling point, or heat of the liquid,

x = quality of wet steam (decimal number),

$\bar{c}_p$ = mean specific heat of superheated steam,

t_s = temperature of superheated steam degrees F.,

t = temperature of the boiling point.

Then

$$(203) \quad H = r + h,$$

$$(204) \quad H_w = xr + h,$$

¹ See A. S. M. E., *Code on Definitions and Values*, 1923, p. 11.

and

$$(205) \quad H_s = r + h + \bar{c}_p(t_s - t) = H + \bar{c}_p(t_s - t).$$

The values of H_s , H , h , and r may be found directly from vapor tables, but the value of H_w must always be computed for the given quality.

82. The Steam Dome on the PV-Plane. The area under the curve *ockdm*, Fig. 26, is the steam dome and represents the region of wet vapors. Anywhere on the curve *ocke* or to the left of it, the substance is in the liquid state. Any point in the curve *kdm* represents dry and saturated vapor. Any point above or to the right of the curve *ekm* represents superheated vapor. When the superheat becomes high enough so that the point representing the state of the substance lies on an isothermal curve without inflection above *k*, as curve *uxy*, the vapor has the characteristics of a true gas. Figure 26 represents, on the

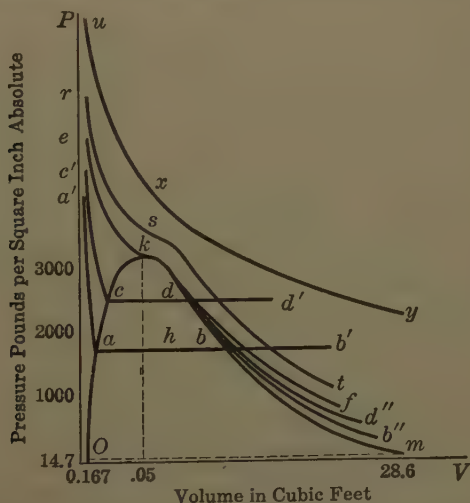


FIG. 26.—THE STEAM DOME ON THE PV-PLANE

PV-plane, the different phases passed through by a substance when transformed from a liquid at or near its freezing temperature into a highly superheated vapor or true gas. It should be noticed that Fig. 26 is somewhat distorted because the volume scale for the curve *ock* is much larger than the scale for curve *kdm*.

The lines $c'cdd''$, ekf , and uxy represent isothermal changes. It is thus seen that when a substance is transformed by addition of heat energy from a liquid into a gas, the isothermal path is broken into three branches, viz., $c'c$, cd , and dd'' . Quality of a wet vapor is represented on this diagram by the ratio ah/ab .

83. Critical Point. The state of the substance represented by point k , Fig. 26, is of special interest. This is known as the critical point. The critical temperature is the temperature above which a substance cannot be liquified by increasing the pressure. At that point the values of all the cardinal properties are called critical values, namely: critical pressure, critical temperature, critical volume, critical entropy, and critical internal energy. Necessarily at the critical point, there is only one value for each of the cardinal properties. The critical point exists for all substances, although there are many for which it has never been determined.

Values of critical temperatures and pressures for a few substances are given in Table 15. The substances are arranged in order of the ascending values of critical temperature. In a general way it may be inferred that the substance having the lowest critical temperature is presumably the most nearly a perfect gas at normal atmospheric temperature and pressure. Helium and hydrogen are at the top in Table 15, and water (H_2O) at the bottom. With the exception of methane, the pressure values fall into an ascending series.

TABLE 15¹
CRITICAL TEMPERATURE AND PRESSURES

SUBSTANCE	t_k DEG. FAH.	p_k LBS. PER SQ. IN. ABS.	SUBSTANCE	t_k DEG. FAH.	p_k LBS. PER SQ. IN. ABS.
Helium.....	-450.0	32.2	Oxygen.....	-181.8	731
Hydrogen.....	-399.8	188.0	Methane.....	-116.5	674
Nitrogen.....	-232.8	492	Carbon dioxide.....	87.8	1071
Carbon monoxide...	-217.7	508	Ammonia.....	270.3	1640
Argon.....	-187.6	710	Water vapor.....	706.1	3226

¹ Values furnished by the Director of the U. S. Bureau of Standards, except the values for water.

84. The Specific Heats of Superheated Steam (H_2O Vapor).

The specific heat of superheated steam varies with its pressure and with the temperature. The magnitude of the variations is very much greater than for gases. Hence it is important to have accurate information regarding this value. Many extensive and intensive researches have been made upon the value of the specific heat of steam. A most exhaustive research which is being carried out by a committee of the American Society of Mechanical Engineers on *Properties of Steam and the Extension of the Steam Tables*,¹ is nearing completion. It has been found that for pressures above 250 pounds the values given in the tables in use hitherto are appreciably in error.

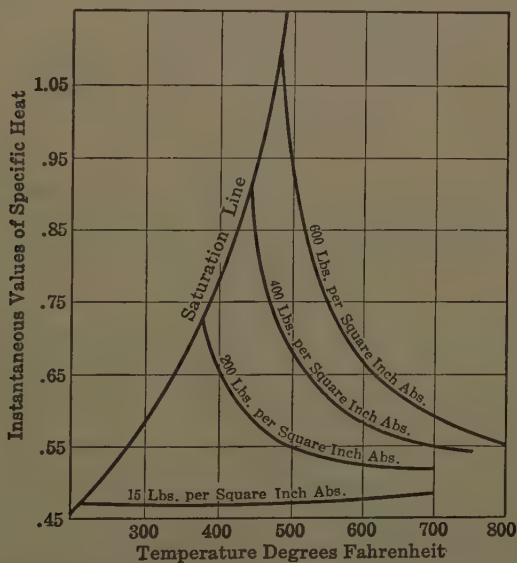


FIG. 27.—INSTANTANEOUS SPECIFIC HEATS OF SUPERHEATED STEAM

Figures 27 and 28 are curves for the specific heat of superheated steam computed from data published in preliminary reports of the steam-table research mentioned above. Figure 27 contains curves showing the variations of the values of the instantaneous specific heat with temperature at three different pressures. Figure 28 shows the variations of mean specific heat values for

¹ See Mechanical Engineering, Feb., 1924, Feb., 1926, Feb., 1929, and Feb., 1930.

the pressure curves of Fig. 27. The curves of Fig. 28 are the integral curves for those shown in Fig. 27. The values of the *mean specific heat* must be used in computing heat quantities for finite changes in temperature taking place at constant pressure.

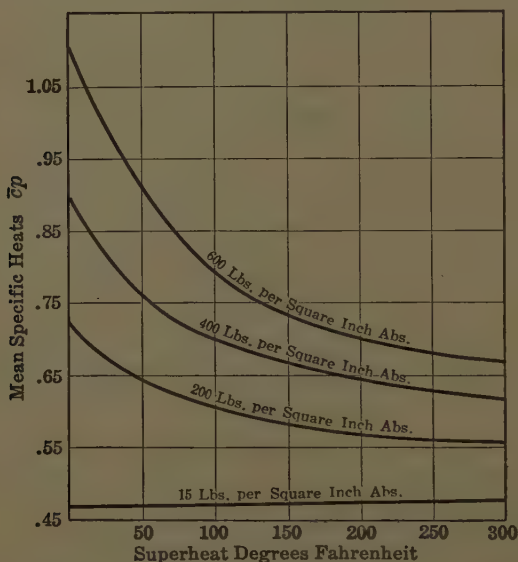


FIG. 28.—MEAN SPECIFIC HEATS OF SUPERHEATED STEAM

85. Entropy. There is no simple definition which is accurate and complete for the cardinal property called entropy. But one which may serve as a beginning is as follows. Entropy is an abstract number or ratio which multiplied by the mean absolute temperature will give the total heat content of a system above the chosen temperature datum. If we take the entropy values as the abscissas of a figure whose ordinates are absolute temperatures, the areas upon such a plane represent, to scale, quantities of heat in a way similar to work quantities on the PV -plane. Thus, entropy may be considered to have the same relation to heat quantities that volumes have to work quantities.

It is only by the use of the calculus that a complete, concise, and satisfactory understanding of the cardinal property, entropy, may be obtained. By (69), p. 60, we have

$$(206) \quad dQ = TdN = dH + dI + AdW.$$

Then

$$(207) \quad dN = \frac{dQ}{T} = \frac{dH + dI + AdW}{T}$$

and

$$(208) \quad N = \int \frac{dQ}{T} = c \int \frac{dT}{T}.$$

Therefore

$$(209) \quad N = c \log_e T + A.$$

In equation (209), A is the constant of integration and c represents specific heat when there is no change of state.

Equation (209) shows that the absolute unit of entropy involves an unknown constant of integration.¹ But when the change of entropy between two temperature levels is determined by the use of a definite integral, the constant of integration cancels out and the change in entropy is readily computed. Thus

$$(210) \quad N_2 - N_1 = \int_{T_1}^{T_2} \frac{dQ}{T}.$$

By substituting (48), page 52, equation (210) becomes

$$(211) \quad N_2 - N_1 = \int_{T_1}^{T_2} \frac{cMdT}{T} = cM \log_e \frac{T_2}{T_1}.$$

Equation (211) is the equation of a logarithmic curve. It is this form which is solved by a step-by-step method in computing entropy of the liquid and entropy of superheated vapor in the various vapor tables used by engineers.

Equation (211) is true provided c (the instantaneous specific heat) is a constant. The value of the specific heat is not exactly constant for any substance; hence, to be sufficiently accurate, it is generally necessary to use the function of T representing c . In that case (211) becomes

$$(212) \quad N_2 - N_1 = M \int_{T_1}^{T_2} \frac{(A \pm BT \pm ET^2 \pm FT^3)dT}{T}.$$

(See equation (55), page 53.)

¹ By taking for the unit of heat energy that which is possessed by a single molecule of a perfect gas it has been possible to compute, by methods of molecular physics, the value of one absolute unit of entropy with a considerable degree of accuracy. See Max Planck, *Origin and development of the quantum theory*. Translated by H. T. Clarke and L. Silberstein, 1920; and E. Wagner, *Annalen der Physik*, vol. 57 (1918), p. 467.

Since all of the latent heat of evaporation, r , is added at constant temperature, the change in entropy during the evaporation of one pound of any substance is represented by the equation

$$(213) \quad N_e = \frac{r}{T}.$$

The total entropy of one pound of dry and saturated steam equals the change in the entropy of the liquid plus the entropy of evaporation. Or

$$(214) \quad N_s = N_1 + N_e = c \log_e \frac{T_2}{T_1} + \frac{r}{T_1};$$

in which

T_1 = temperature datum, degrees F. abs.,

T_2 = temperature of the boiling pt. degrees F. abs.,

r = latent heat of evaporation in B.t.u.,

c = instantaneous specific heat of the liquid,

N_e = entropy change during evaporation of one lb. of steam,

N_1 = entropy change for one lb. of the liquid,

N_s = total change in entropy during generation of one lb. of dry and saturated steam.

The following summarized statements regarding entropy based upon equations (206) to (214) inclusive may now be made.

1. Entropy, like energy and volume, varies with its mass.
2. Entropies like energies may be added. If a system consists of a substance in the liquid and vapor phases, the total entropy of the system is the sum of the entropy of the liquid and the entropy of the vapor determined separately.
3. If the heat unit is the B.t.u. and the temperature unit an absolute degree Fahrenheit, the entropy unit is B.t.u. per degree for one pound of the substance. This unit has never been given an official name.
4. Entropy is a ratio and in the integral form contains an unknown constant of integration. But in the form of a definite integral the constant of integration cancels out. For steam-table work, entropy of the liquid at 32° F. is arbitrarily assumed

to be zero. However, any other convenient datum could be chosen.

5. The entropy of a substance or a system, made up of a liquid and its vapor, has a single fixed value for every state or condition of that system. The value of the entropy depends directly upon and is fixed by that state. If an ideal system passes from a given state through a succession of states along a series of reversible paths, finally returning to the initial state, the value of the entropy passes through a corresponding succession of values, and also returns to its initial value at the end of the cycle. Hence entropy is a function of the state only, and is thus properly called a cardinal property.

86. The Steam Dome on the TN -Plane. The evaporation of any liquid involves the addition of heat along a definite path. The path upon the temperature-entropy plane illustrates what takes place, and makes very simple some relations which are not so apparent when other planes are employed. The ordinates of the TN -plane are always temperatures in degrees absolute. Abscissæ are always values of entropy. Values of entropy of the liquid, entropy of evaporation, and their sum, or total entropy, are usually given as separate items in all vapor tables. But the various values of entropy have to be computed for gases.

The line $ac'c''c'''$, Fig. 29, shows the path followed when one pound of water is heated from 32° F., equal to 491.6° F. absolute, to the boiling point, at constant pressure, and evaporated into steam and superheated at the same constant pressure. This is the program followed in steam formation in a steam boiler. The area between the line $ac'c''c'''$ and the x -axis represents to scale the amount of heat absorbed. The line $ac'k$ represents the liquid line. Points in the liquid line represent the entropy of one pound of water at a succession of boiling point temperatures from 491.6° F. absolute up to a temperature of 1165.6° F. absolute. The value of the entropy of the liquid at 491.6° F. (the ice point) is considered to be zero. Consider a pound of water in a boiler in which the pressure is held constant at P_c pounds per square foot absolute. The corresponding boiling point tem-

perature is T_c °F. abs. In Fig. 29 the horizontal distance cc' represents the change in entropy as the temperature of the water was raised from 491.6° F. abs. up to temperature T_c . At T_c the temperature remains constant as more heat is added. The entropy now increases at constant temperature along the horizontal line $c'c''$ until the liquid has all changed to vapor and the total

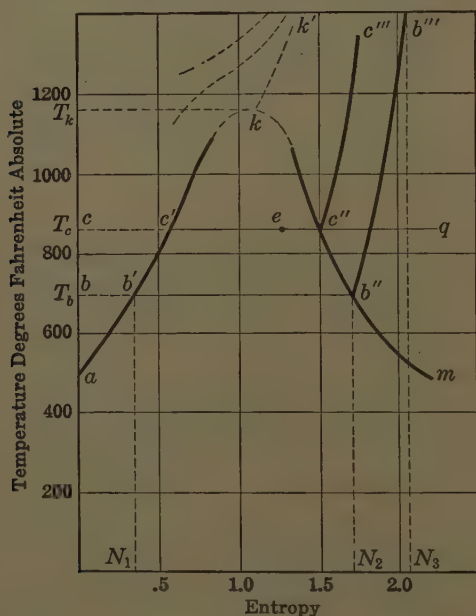


FIG. 29.—THE STEAM DOME ON THE TN -PLANE

value of the entropy has now become that at c'' . The total change in entropy from water at the ice point to just dry and saturated vapor is represented to scale by the length of the line cc'' . As more heat is added, the temperature begins to rise again showing that the steam is being superheated. This change is represented on the TN -plane by a second logarithmic curve $c''c'''$. So long as heat addition continues, this curve will continue upward until the temperature of dissociation is reached. The curve $kc''b''$ represents the values of entropy of dry and saturated steam at a succession of decreasing temperatures. At any point within the dome, $ac'kb''m$, the substance is a mixture of liquid and

vapor. If the evaporation process should be stopped at any point as e on the line $c'e''$, the quality of the mixture is equal to the length $c'e \div c'e''$, both lengths being expressed in entropy units.

87. Isothermal and Adiabatic Curves on the TN -Plane. The isothermal curve indicates changes in pressure, volume, entropy, and internal energy, while the temperature remains constant by proper control. Hence on the TN -plane an isothermal change is indicated by a straight line parallel to the x -axis, as the line $cc''q$, Fig. 29. An adiabatic change is one which takes place without flow of heat energy into or away from the substance. Any external work which is done during an expansion must be at the expense of energy already stored in the substance. This necessitates a drop in temperature or a partial condensation of the substance or both. During an adiabatic change, there can be no change in the entropy, for

$$(215) \qquad \Delta N = \frac{\Delta Q}{\bar{T}},$$

where

ΔN = finite entropy change,

ΔQ = finite heat change,

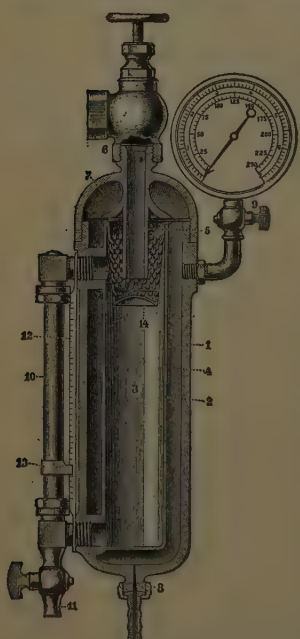
$\bar{T}$ = mean temperature during heat change Q .

If, in equation (215), ΔQ is zero, ΔN must be zero also. Hence an adiabatic change on the TN -plane must be represented by a straight vertical line, as $b'''N_3$, Fig. 29.¹

88. Measurement of Quality. Special instruments called steam calorimeters are employed to determine the quality of wet steam. A small representative sample of wet steam is collected from the steam pipe and passed through the calorimeter. From the observations made the quality may be computed, as will be explained below. Care must be exercised in order to obtain a representative sample. A sampling tube, Fig. 31, is inserted perpendicularly to the axis of a vertical section of the steam main. The sample should not be taken from a long horizontal pipe,

¹ See page 347 for further discussion on the topic of entropy.

because the water, being much denser than the vapor, tends to separate out and settle to the lower side of the pipe. In that condition it is difficult or impossible to obtain an average sample which is truly representative of the main body of steam flowing in the pipe. Because of eddies and inertia effects, the sample should be taken at least ten pipe-diameters downstream from any sharp bends or other obstructions. The sampling tube should be a half-inch pipe closed off at its inner end. It should be long enough to reach nearly across the steam main and contain at least twenty 1/8-inch perforations arranged in a spiral along the pipe. The perforations should not be closer than one-half inch to the walls of the main.



(American Schaeffer & Budenberg Corp.)

FIG. 30.—CARPENTER
SEPARATING CALORIMETER

89. Carpenter Separating Calorimeter. There are two types of calorimeters which depend for their operation upon two different fundamental principles. They are the separating calorimeter and the throttling calorimeter. In Fig. 30 is shown a sectional view of a Carpenter separating calorimeter. It is so constructed that the greater inertia of the liquid is employed to cause mechanical separation of the liquid and vapor. The mixture of dry steam and moisture flows into a perforated cup within the calorimeter at a fairly high velocity. The path of flow is suddenly reversed in the cup. The moisture, unable to make the sudden change in

direction of flow due to its greater inertia, is nearly all thrown out of the moving body of steam. It then falls through the perforated cup into a well where it may be measured by a scale placed along side of a sight glass. The dry steam leaving the calorimeter through a discharge orifice may be condensed and weighed; or the orifice

may be used as a steam meter to measure it. The passages for the steam as it approaches the exit orifice serve as a jacket to reduce radiation losses from the calorimeter to a minimum. When the weight of water caught and the weight of dry steam flowing per unit of time are known, the quality is computed as follows:

$$(216) \quad x = \frac{M + R}{M + m}.$$

Here

x = quality of the wet steam (a decimal fraction),

M = lbs. of dry steam escaping from the calorimeter orifice per min. or per hr.,

m = lbs. of liquid caught per min. or per hr.,

R = lbs. of water condensed per min. or per hr. due to radiation from the calorimeter.

The value of R is usually so small as to be negligible unless extreme precision is desired.

It is not always convenient to condense and weigh the steam. It may be computed very readily by Napier's formula. During the years 1865-1868, Robert Napier, a Scottish ship-builder and engineer, carried out many experiments on the flow of steam through orifices from a high-pressure region into the air. Professor William J. Rankine¹ devised the following formula for orifices having well-rounded entrances, based upon Napier's experimental work.

$$(217) \quad M = \frac{pa}{70},$$

Where

a = area of the orifice in sq. ins.,

M = lbs. of dry steam flowing per sec.,

p = pressure above the orifice in lbs. per sq. in. absolute,

p_2 = pressure in the discharge region in lbs. per sq. in. absolute.

Equation (217) is now known as *Napier's formula* for the flow of steam. It is an empirical formula being based entirely upon

¹ See Engineer (London), Nov. and Dec., 1869, pp. 352 and 358.

experiment. It is good only so long as p_2 is less than $0.58 p_1$. The results from the formula are not absolutely correct but may be in error 2 or 3 per cent. However, the results are sufficiently accurate for most calorimeter work.

The calorimeter is generally fitted with a pressure gage having a double scale. One scale is graduated to give pressures in pounds as usual. The other reads in pounds of steam flowing in ten minutes.

The mechanical separation of the moisture from the steam is not perfect. The efficiency of separation is probably about 98 per cent in most instances. The chief advantage possessed by the separating calorimeter is that large percentages of moisture may be measured by its use, perhaps as much as 30 per cent of water. But in these days of highly superheated steam, a calorimeter possessing this ability is seldom required.

90. Throttling Calorimeters. The throttling calorimeter was invented and first used by Professor C. H. Peabody¹ in 1888. Before that time calorimeters were clumsy and inaccurate instruments. Professor R. C. Carpenter developed the improved design shown in Fig. 31. It consists of an enlarged chamber in the path of the flowing steam, thickly covered with heat-insulating material. A deep thermometer cup is provided in the chamber for insertion of a thermometer. By means of a mercury manometer connected to the calorimeter chamber, the steam pressure in the chamber is measured. This pressure is ordinarily from one to four inches of mercury above the pressure of the atmosphere.

The throttling-calorimeter action is a modified Joule experiment so arranged that it is a continuous process. A large and sudden drop in pressure occurs as the steam flows from the main into the calorimeter chamber through the inlet orifice. This pressure drop produces a high velocity of flow. But the kinetic energy of flow is reconverted into heat by eddies and friction within the calorimeter chamber where the velocity is reduced to a negligible value. The steam then escapes into the atmosphere

¹ See Transactions A. S. M. E., vol. 9.

at relatively low velocity through the calorimeter exit. The steam expansion through the orifice into the calorimeter chamber takes place without doing any external work and no heat energy is received from or given up to an external source. *This is known as the throttling process. It is a constant heat change with no external work being done.*

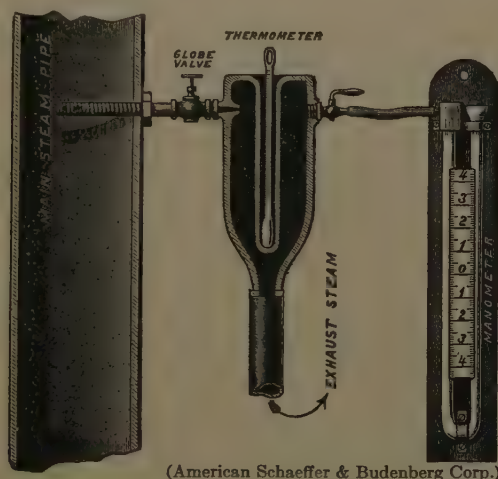


FIG. 31.—CARPENTER THROTTLING CALORIMETER

By referring to the steam tables, it will be observed that at 125 pounds per square inch absolute the total heat content of one pound of dry and saturated steam above 32° F. is about 1190 B.t.u. At a pressure of 15 pounds per square inch absolute, one pound of dry and saturated steam contains 1150 B.t.u. Thus when steam is throttled from 125 down to 15 pounds pressure with no heat lost or gained and no external work done during the change, the 40 B.t.u. which are made available must do one of two things or both together, namely: superheat the steam, or evaporate the moisture if the steam be initially wet. Referring to the steam table again, it is observed that at 15 pounds pressure about 970 B.t.u. of latent heat are required to evaporate one pound of water from the boiling point into dry and saturated steam, and

$$\frac{40}{970} = 0.041.$$

Therefore if the wet steam at 125 pounds pressure in the main carries more than about 4 per cent of moisture, that part of the moisture in excess of 4 per cent would pass through the calorimeter unevaporated. However, if the initial moisture content were less than 4 per cent, it would all be evaporated and the steam in the calorimeter chamber would become superheated to some degree. The latter condition must obtain if useful data from which the initial quality of the wet steam in the steam main can be computed is made available. The total heat content of the steam in the steam main and in the calorimeter chamber is the same. This fact furnishes the condition upon which two equivalent algebraic expressions may be stated. The solution of this heat equation will then yield the value of the quality in the steam main.

Let

$\bar{c}_p$ = mean specific heat of superheated steam,

h_1 = heat of the liquid in 1 lb. of water above 32° F. at pressure p_1 ,

H = total heat above 32° F. in 1 lb. of dry and saturated steam at pressure p_1 ,

H_c = total heat above 32° F. in 1 lb. of superheated steam in the calorimeter chamber at pressure p_c .

H_m = total heat above 32° F. in 1 lb. of wet steam at pressure p_1 , in the steam main,

m = weight of moisture in 1 lb. of wet steam,

p_1 = pressure in lbs. per sq. in. abs. in the steam main,

p_c = pressure in lbs. per sq. in. abs. in the calorimeter chamber,

r_1 = latent heat for 1 lb. of dry and saturated steam at pressure p_1 ,

t = temperature of the boiling pt. degrees F., in the calorimeter chamber at pressure p_c ,

t_c = temperature as measured in the calorimeter chamber, degrees F.,

x_1 = quality of steam in the main at pressure p_1 (a decimal number).

Then

$$(218) \quad H_w = x_1 r_1 + h_1.$$

For the conditions in the steam main at pressure p_1 , equation (218) represents the heat content of one pound of wet steam. For the conditions in the calorimeter chamber, the relations are

$$(219) \quad H_c = H + \bar{c}_p(t_c - t).$$

But since the throttling process is a constant heat change,

$$(220) \quad H_w = H_c,$$

hence

$$(221) \quad x_1 r_1 + h_1 = H + \bar{c}_p(t_c - t).$$

The value of each item is known except x_1 which must be found by solving (221). The result¹ is

$$(222) \quad x_1 = \frac{H + \bar{c}_p(t_c - t) - h_1}{r_1},$$

and

$$(223) \quad m = 1.00 - x_1.$$

All the quantities in the left-hand member of equation (221) are at steam-main pressure p_1 . All the quantities in the right-hand member are for the calorimeter pressure p_c , which is never very far from 15 pounds per square inch absolute.

There are certain limits outside of which the indications from a throttling calorimeter are worthless for computing the quality. Of course instruments indicating correct pressures and temperatures must be employed. In order to be certain that all the moisture in the wet steam has been evaporated in the calorimeter chamber, the temperature difference $(t_c - t)$ should be not less than 10 degrees. Hence for different initial steam pressures p_1 , the minimum quality which is measurable varies. The lower limits for various pressures in the main are approximately as given in Table 16.

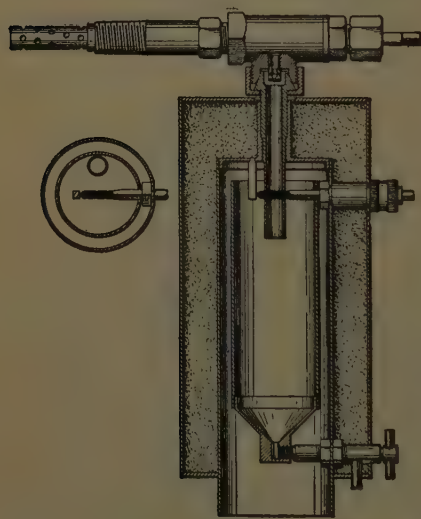
¹ This equation may be solved rapidly and accurately by use of a nomograph by C. A. Kulmann, Power, Dec. 13, 1927, p. 933.

TABLE 16
MINIMUM QUALITY BY THROTTLING CALORIMETER

STEAM PRESSURE LBS. PER SQ. IN. ABS.	MINIMUM QUALITY	STEAM PRESSURE LBS. PER SQ. IN. ABS.	MINIMUM QUALITY
300	0.940	100	0.965
200	0.950	50	0.978
150	0.955		

If the throttling calorimeter is heavily insulated and accurate instruments are used, the values of quality obtained by its use have a high degree of accuracy.

The universal calorimeter is a combination of the throttling calorimeter with a separating calorimeter. The steam usually enters the throttling section directly from the main, and then it passes on into the separating part.



(Ellison Draft Gage Co.)

FIG. 32.—ELLISON U-PATH STEAM
CALORIMETER

The Ellison U-Path Calorimeter, views of which are shown in Fig. 32, is an improved modern design of this type. The steam enters the instrument horizontally with no up-turns between the sampling tube and the orifice plug, in passing through which the steam is throttled to atmospheric pressure. Immediately below the orifice, the steam impinges on a plane surface, thereby

being divided into four streams which meet and mix again below the plug. The low pressure steam flows in a U-path down one side of the calorimeter chamber and up the other, escaping at the top, where it turns downward through an annular passage and escapes to the atmosphere. This annular passage serves as a steam jacket for the calorimeter chamber. The instrument is further

insulated against the loss of heat by being enclosed in a case of polished metal which contains lampblack, an efficient non-conductor of heat.

Any momentary abnormal moisture content in excess of that due to the throttling process is separated in the steam chamber and re-evaporated by the superheated steam, thus giving a true mean superheat temperature at the calorimeter thermometer.

The U-path forms an efficient means of separating the water in a mixture of water and dry steam in its expanded low-pressure state. The instrument thus forms a combined throttling, separating, and re-evaporating calorimeter of high efficiency.

Unless the steam contains too much moisture to be measured by the throttling part, the separator part does not function. For very wet steam, however, the separator section makes it possible to find the quality of the wet steam leaving the throttling section, and consequently the quality in the main as follows:

$$(224) \quad x_1 r_1 + h_1 = x_c r_c + h_c.$$

The subscript 1 in equation (224) refers to values at steam-main pressure p_1 and the subscript c refers to values at the pressure p_c in the calorimeter chamber. The value of x_c may be computed from the separating-calorimeter readings and formula (216), page 127. Then all the quantities in equation (224) are known except x_1 . The solution of (224) gives

$$(225) \quad x_1 = \frac{x_c r_c + h_c - h_1}{r_1}.$$

ILLUSTRATIVE PROBLEM

Given the following data from a throttling calorimeter from which to compute the quality and the moisture: $p_1 = 125$ lbs. per sq. in. gage; $t_c = 225^\circ$ F.; $p_c = 2$ in. Hg. above atmospheric; barometer reading = 29.75 in. Hg.

Then

$$p_1 = 125 + (0.491 \times 29.75) = 139.6 \text{ lbs. per sq. in. abs.}$$

$$p_c = 0.491(29.75 + 2.00) = 15.6 \text{ lbs. per sq. in. abs.}$$

For $p_1 = 139.6$ it is found from steam tables that

$$h_1 = 324.5 \text{ B.t.u.,} \quad r_1 = 867.9 \text{ B.t.u.}$$

For pressure $p_c = 15.6$ lbs. per sq. in. abs., it is found by consulting steam tables that $H = 1151.4$ B.t.u., and the corresponding boiling point is $t = 215^\circ$ F. It is also seen by reference to Fig. 28, and a study of steam-table data will confirm it, that the value of $\bar{c}_p$ at a pressure in the neighborhood of 15 lbs. per sq. in. abs. is approximately 0.48 for all degrees of superheat.

The value of the quality may now be obtained by substituting the above values in equation (222) and solving. The result is

$$x_1 = \frac{1151.4 + 0.48(225 - 215) - 324.5}{867.9} = 0.958; = 95.8 \text{ per cent.}$$

Then

$$m = 1.00 - .958 = .042 \text{ or } 4.2 \text{ per cent.}$$

PROBLEMS

It is intended that the following problems shall be solved without the aid of the charts which usually accompany steam tables.

1. Plot upon engineers' cross-section paper and to a convenient scale the following curves, taking the data from steam tables. Five or six points for each curve will be sufficient. Plot between 32° F. and 706° F. as temperature limits and between pressure limits of 0.0885 and 3226 lbs. per sq. in. abs.

CURVE NO.	ORDINATES	ABSCISSAS
1	Temperatures, $^\circ$ F.	Pressure, lbs. per sq. in. abs.,
2	Temperatures, $^\circ$ F.	Entropy of the liquid,
3	Temperatures, $^\circ$ F.	Total entropy of the steam,
4	Specific vol. in cu. ft.	Pressure, lbs. per sq. in. abs.,
5	Total heat in B.t.u.	Pressure, lbs. per sq. in. abs.
6	Latent heat in B.t.u.	Pressure, lbs. per sq. in. abs.

2. Compute values of $\bar{c}_p$ from the superheated steam table for the following absolute pressures in lbs. per sq. in.: 15; 100; 200; 400; and 600; and at each pressure for the following ranges of superheat in degrees F., beginning in each case at 0° of superheat: 50; 100; 200; and 400. Arrange the results in a convenient table.

3. Compute the quality of wet steam from the following separating calorimeter data: neglect radiation loss; diameter of calorimeter-exit orifice = $5/32$ inch; weight of water caught per hour = 9.5 lbs.; $p_1 = 135$ lbs. per sq. in. abs.

4. Plot the steam dome on the PV -plane, taking the necessary data from the steam table.

5. From steam-table data plot the steam dome on the TN -plane.

6. Compute the quality from the following throttling-calorimeter data: $p_1 = 160$ lbs. per sq. in. gage; $p_c = 5$ in. Hg. above the pressure of the atmosphere; $t_c = 240^\circ$ F.; barometer reading = 29.35 in. Hg.

7. Compute the maximum amount of moisture which may be measured with certainty by a throttling calorimeter for the conditions in Problem 6.

8. When the total entropy of a pound of a mixture of water and vapor is 1.5100 at 200 lbs. per sq. in. abs., what is the quality?

9. Compute the total B.t.u. above 32° F. supplied to a steam engine per horsepower hour for the following conditions: pounds of steam supplied per horsepower hr. = 17.5; $p_1 = 160$ lbs. per sq. in. gage; $x_1 = 0.95$; barometer reading = 29.00 in. Hg; exhaust pressure = 2 lbs. per sq. in. abs.

10. What would be the B.t.u. supplied per horsepower hour if the steam were initially superheated 150° F., all other conditions remaining the same as in Problem 9?

11. Compute the increase in entropy when one pound of liquid H_2O is heated from 75° F. to 400° F. Assume the mean specific heat for that temperature range to be 1.03. Check the result by comparing with the value taken from the steam tables.

12. Compute the change in entropy when 7.5 lbs. of steam at a pressure of 150 lbs. per sq. in. abs. is heated at constant pressure from 400° F. to 760° F. (Find the value of $\bar{c}_p$ from the superheated-steam table.)

13. Five pounds of wet steam have a total entropy of 7.250 at a pressure of 150 lbs. per sq. in. gage when the barometer reading = 28.90 in. of Hg. Compute the quality.

14. Compute the quality of wet steam from the following universal-calorimeter data: $p_1 = 160$ lbs. per sq. in. gage; $p_c = 1.00$ in. Hg. above atmospheric; barometer reading = 29.45 in. Hg.; diam. of the calorimeter orifice is 1/8 in.; water caught in the separator = 2.00 lbs. per hr. Neglect radiation loss.

15. Compute the quality when 1 lb. of wet steam contains 1085 B.t.u. above 32° F. at a pressure of 200 lbs. per sq. in. gage. The barometer reading is 29.10 in. of Hg.

16. Compute the total external work in B.t.u. done when 1 lb. of H_2O in the liquid state at 75° F. is changed into dry and saturated steam at a pressure of 175 lbs. per sq. in. abs.

17. Find the final pressure if superheated steam at an initial pressure of 250 lbs. per sq. in. abs. and $t_1 = 601^\circ$ F. expands adiabatically until it becomes just dry and saturated. If the expansion follows the law $PV^k = K$, find the value of k . (NOTE.—Construct a TN -diagram.)

18. Find the final pressure when initially superheated steam at a pressure $p_1 = 275$ lbs. per sq. in. gage and 200° F. of superheat expands adiabatically until the quality becomes 0.94. (Use the TN -diagram of Problem 17.)

19. One pound of initially dry and saturated steam expands from a pressure of 300 lbs. per sq. in. gage to 2 lbs. per sq. in. abs., doing external work. Sufficient heat is supplied as the expansion proceeds to maintain just dry and saturated steam throughout the expansion period. Compute the B.t.u. required.

20. Compute the change of entropy when 5 lbs. of air are heated from 75° F. to 600° F. at constant volume; at constant pressure. (Use the equations for specific heat given in Table 10, page 72 herein.)

21. If the latent heat of evaporation for one pound of steam at a pressure 180 lbs. per sq. in. gage is 850.8 B.t.u., compute the entropy of evaporation.

22. The temperature of 500 pounds of water is to be increased from 70° F. to 190° F. by condensing in it exhaust steam from an engine. The pressure of the exhaust steam is 15 lbs. per sq. in. abs. and its quality is 0.85. How many pounds of exhaust steam will be needed? (Assume the mean specific heat of water for this temperature range to be 1.01.)

23. One pound of wet steam expands adiabatically from $p_1 = 250$ lbs. per sq. in. gage and $x_1 = .93$ to $p_2 = 1$ lb. per sq. in. abs. Using the steam tables only, find the value of x_2 and the value of k for the adiabatic equation $PV^k = K$.

24. One hundred B.t.u. are imparted to one pound of liquid H_2O whose initial temperature is 65° F. Compute the increase in entropy if $\bar{c}_p = 1.02$.

25. What is the difference between specific heat and entropy? Discuss by use of equations.

CHAPTER XI

ENGINE CYCLES, SECOND LAW OF THERMODYNAMICS, AND THERMODYNAMIC SCALE OF TEMPERATURE

91. Reversible and Irreversible Cycles. A thermodynamic cycle (see page 85) is said to be reversible when it satisfies the following conditions. First, when a substance or a system composed of several substances can be made to retrace in reversed order a connected series of transformation steps and each step is retraced in reversed manner; secondly, when the external reactions are the same as in the direct cycle but reversed step by step; thirdly, at the end of the retraced cycle, the system itself and all connected systems must be restored to the initial state which existed before the direct cycle started.

There are three cases of irreversibility which are of especial interest in the study of heat engines.

First, the conversion of mechanical work into heat energy by friction. Consider Davy's experiment of melting ice by rubbing two blocks of it together. The mechanical energy imparted to the molecules causes their velocity of vibration to increase materially, thus changing the water from the solid to the liquid state. Or, consider the heat generated by friction in the bearings of a steam turbine; no possible way has ever yet been devised to reverse this action and derive useful work from the bearing by cooling it. The same is true of friction produced by water flowing in a pipe line. In general, the transformation of work into heat by any kind of friction is an irreversible process.

Second, heat carried away by conduction through some body from a high-temperature region to a low-temperature region is also an irreversible process. For instance, no process has ever been discovered whereby it would be possible to cause heat to flow, of its own accord, from the relatively cool water in a steam boiler out through the boiler surfaces into the hot furnace gases.

Third, throttling the flow of steam from a high-pressure region through an orifice to a low-pressure region without performance of external work is an irreversible process. An excellent example of irreversible processes is the throttling calorimeter. It is very evident that the fluid cannot, unaided, flow back from the low-pressure to the high-pressure region. But it may be compressed by doing work upon it, and thus be forced to return. However, carrying out such a program necessitates expenditure of mechanical work and a consequent change in external systems.

In our practical everyday world there are, in the strictest sense, no truly reversible cycles. Frictionless motion cannot be produced. Radiation of heat energy from a hot body to a cold body cannot be entirely checked. So far as is known, there is no substance which is a perfect non-conductor of heat. There is no perfect gas. But we can imagine some of the things which would happen if ideal substances and ideal conditions were available, and develop equations representing ideal thermodynamic changes. There are a number of ideal changes which may be made parts of an ideal reversible cycle.

92. Ideal Heat Engine. An ideal mechanism which is operated by an ideal substance passing through a series of ideal reversible changes in an ideal reversible cycle would constitute a perfect or ideal heat engine. It will be shown that such an ideal engine would be able to convert heat energy into mechanical work with the highest possible efficiency. The high efficiency of an ideal engine is not and never will be completely realized in practice. However, it is very helpful to study the features of an ideal engine and its performance characteristics as a standard of perfection. By so doing, it becomes possible to construct and operate actual engines which attain efficiencies approaching as nearly as may be those of the ideal.

The fundamental principle upon which ideal and actual engines operate is the same. *They both receive a supply of heat energy from a source at high temperature, transform a part of the energy into useful work, and discharge the remainder of it to a low-temperature region.*

93. Sadi Carnot. Sadi Nicolas Leonhard Carnot (1796–1832) was the elder son of Lazare Nicolas Marguerite Carnot. Lazare Carnot was a mathematician of note, minister of war, and statesman during the French Revolution under Napoleon. The success of several of Napoleon's campaigns was due largely to the great organizing ability of Carnot. His son Sadi was trained for a military career. He had a decided gift for mathematics and natural philosophy. He became an engineer in the army, but found the menial duties of the army to be very distasteful, and felt hampered in his scientific studies, for which he possessed abilities of the very highest order. After being made a captain of engineers in 1828, he resigned. From then until his untimely death at the age of 36 years, he pursued work in his chosen field. It was in 1824 that his one and only published paper appeared. It was a small pamphlet, *Reflexions sur la Puissance Motrice du Feu*. In it he proposed the first ideal engine and the first ideal cycle of operation. This paper contains a report on only a fragment of his many scientific studies, but it is sufficiently important to perpetuate his name as long as there are heat prime movers. His paper was not widely circulated, and very little attention was given it until 1848, when William Thomson (Lord Kelvin) accidentally came across it and recognized its true worth. That paper and other unpublished writings show that Carnot not only believed in the now accepted dynamic conception of heat energy, but, also that he noted down several of the very methods later developed by Joule for finding the mechanical equivalent of heat. Sadi Carnot was an uncle of Marie François Sadi Carnot, president of the French Republic, 1887–1894. Altogether the three Carnots left an indelible mark upon civilization in the departments of military science, in natural science, and in governmental science.

94. Carnot Cycle and Engine. Carnot was the first to recognize the fundamental principle that any substance being used as a medium to transform heat energy into mechanical work must pass through a succession of changes, finally returning to its initial state; thus traversing a complete cycle. The cycle must be complete if it is to be possible to determine what proportion of the heat supplied is converted into useful work.

Before and since Carnot's time, many inventors have expended much effort trying to find a working substance which would have an efficiency of conversion greater than that of H_2O in its liquid

and vapor states. Most of these efforts have been more or less haphazard experiments without success. Carnot was the first to determine from broad theoretical considerations the right answer to the problem. His answer given in the following development will probably prove to be the correct one for all time. The graphical and mathematical treatment of this problem was first formulated by Clapeyron ¹ in 1834. It has since been somewhat modified by Rudolph Clausius, Lord Kelvin, William Rankine, and others.

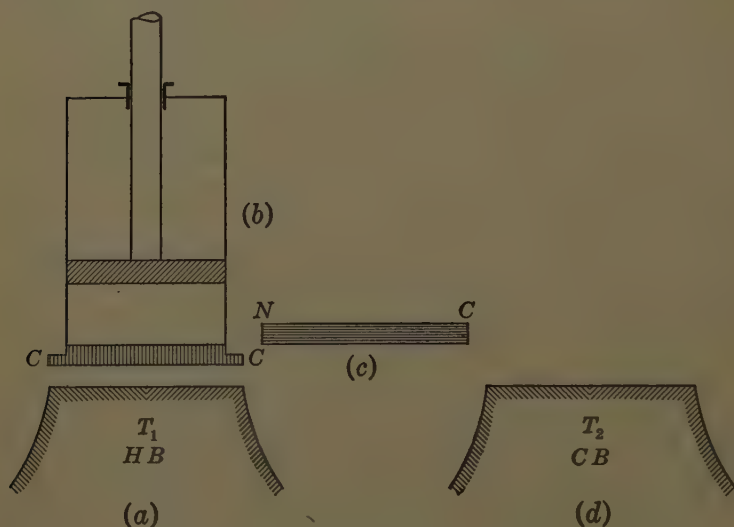


FIG. 33.—CARNOT IDEAL ENGINE

The essential features of Carnot's ideal engine and cycle are:

1. A cylinder and movable piston made of a perfect heat insulator (Fig. 33 *b*), the piston to be frictionless in its movements.
2. Cylinder head CC a perfect conductor of heat.
3. A movable cylinder head NC , a perfect heat insulator.
4. An infinite source of heat HB always at temperature T_1 . (Corresponds to boiler furnace.)
5. An infinite source of cold CB always at temperature T_2 . (Corresponds to the atmosphere or condenser.)

¹ Journal de l'Ecole Polytechnique, vol. 19 (1834).

6. A perfect gas, a medium by means of which heat energy is transformed into mechanical work which is imparted to the movable frictionless piston.

In carrying out the four operations of the Carnot ideal cycle, it is assumed that the weight of the working substance confined in the cylinder is constant throughout the cycle and that it is used over and over as the cycle is repeated.

First Operation. The cylinder containing the gas at pressure P_1 , volume V_1 , and temperature T_1 is placed in thermal communication with the hot body HB through its perfect conducting head CC . Heat flows into the gas at constant temperature T_1 . It is important to note that no matter how much heat energy is absorbed from HB its temperature remains constant at T_1 degrees. The gas expands to volume V_2 and pressure P_2 , forcing the piston outward and performing external mechanical work. But the expansion is so

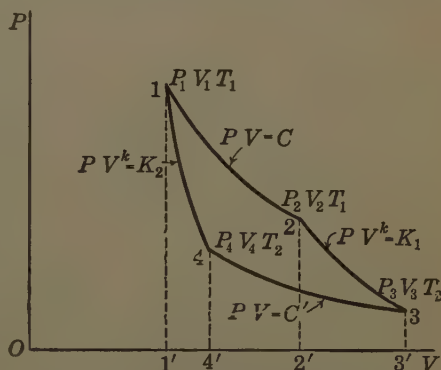


FIG. 34.—THE CARNOT CYCLE ON THE PV -PLANE

controlled that the temperature T_1 of the gas remains constant and equal to the temperature of HB ; hence the expansion is isothermal. The work done, W_1 , in foot-pounds, is represented in Fig. 34 by the area 122'1'. By equations (134) and (136), page 80, the work in foot-pounds is

$$(226) \quad W_1 = P_1 V_1 \log_e r = MRT_1 \log_e r.$$

This amount of work is equivalent to the entire quantity of heat energy withdrawn from HB during the first operation, since the internal energy of the gas remains unchanged as indicated by the constant temperature.

Second Operation. At volume V_2 represented at 2, Fig. 34, the cylinder is removed from the hot body, and the non-conducting

head NC instantly put in place. The gas expansion continues without gain or loss of heat, and the piston performs more external work until the volume becomes V_3 at the temperature T_2 . This change is an adiabatic expansion. The external work W_2 during the second operation, is represented in Fig. 34 by the area 2 3 3' 2'. By equation (127), page 79, the work in foot-pounds equals

$$(227) \quad W_2 = \frac{MR(T_1 - T_2)}{k - 1}.$$

Third Operation. At volume V_3 , the non-conducting cover NC is removed, and the cylinder is instantly placed in thermal contact with the cold body CB at temperature T_2 . The gas is then compressed at constant temperature T_2 ; the piston doing work upon the gas until the volume is reduced to V_4 . This step is an isothermal change. The work done upon the gas by the piston may be considered as negative work. It is represented by the area 344'3'. Analytically, the work in foot-pounds equals

$$(228) \quad W_3 = MRT_2 \log_e r.$$

The heat equivalent of W_3 is transferred from the gas to the cold body. Similar to HB the temperature of CB remains constant at T_2 no matter how much heat energy may be transferred to it.

Fourth Operation. The cylinder is removed from the cold body, and the non-conducting head instantly put on. Compression then continues adiabatically until the gas is restored to its initial condition at P_1 , V_1 , and T_1 . This work, which is also negative work, is represented by the area 411'4'. Its value in foot-pounds is

$$(229) \quad W_4 = \frac{MR(T_1 - T_2)}{k - 1}.$$

In order that the gas may be restored to the initial conditions as to temperature, pressure, and volume by the closing step in the cycle, a certain relation must obtain between the volumes V_1 , V_2 , V_3 , and V_4 , as follows: From the relations shown in Table 12, page 81, it is seen that during adiabatic expansion

$$(230) \quad \frac{T_1}{T_2} = \left(\frac{V_3}{V_2} \right)^{k-1};$$

and that during adiabatic compression

$$(231) \quad \frac{T_1}{T_2} = \left(\frac{V_4}{V_1} \right)^{k-1}.$$

Eliminating T_1/T_2 from equations (230) and (231), one obtains

$$(232) \quad \frac{V_3}{V_2} = \frac{V_4}{V_1} \quad \text{or} \quad \frac{V_2}{V_1} = \frac{V_3}{V_4}.$$

Equation (232) indicates that the ratio of isothermal compression during the third operation must equal the ratio of isothermal expansion during the first operation. Then the cycle will properly close at the end of the fourth operation.

An inspection of equations (227) and (229) shows that $W_2 = W_4$. Since W_2 represents work done by the gas and W_4 work done upon the gas, the net result of these two operations so far as external work is concerned is zero. Then the net external work which is done during a complete cycle is represented by the closed area 1234, from (226) and (228) it equals

$$(233) \quad W_1 - W_3 = MR(T_1 - T_2) \log_e r.$$

Since efficiency in general equals the result accomplished divided by the effort made, the efficiency of the Carnot ideal cycle is

$$(234) \quad e_c = \frac{MR(T_1 - T_2) \log_e r}{MR T_1 \log_e r}.$$

This reduces to

$$(235) \quad e_c = \frac{T_1 - T_2}{T_1}.$$

Equation (235) is a very important one in thermodynamics, for by its terms it is observed that the *value of the efficiency of the ideal Carnot engine depends only upon the absolute temperatures of the hot and cold bodies. The value of the efficiency is independent of the material* used for the working substance or the state in which that substance may exist. But the area of the closed figure in Fig. 34 and consequently the *amount of external work done per cycle does depend upon the kind of working substance used.* The kind of working substance in turn determines, within certain

limits, the necessary size and weight of an engine required to transform a certain amount of heat into work per unit weight of working substance. The advantages and limitations of various substances are discussed in § 205 of this book.

The Carnot Engine and Cycle Are Reversible. There is, in the ideal case, no reason why Carnot's engine cannot be driven by a motor in a reversed direction with the result that each and every operation will be carried out in a reversed manner. In the reversed cycle, mechanical work will be expended upon the engine driving it backwards. By virtue of this expenditure of mechanical energy, heat energy will be taken from the cold body and transferred to the hot body. The quantity of heat energy so transferred back to the hot body will, in the ideal case, exactly equal the quantity of heat energy taken away in the direct cycle. The reversed engine thus becomes a heat pump, pumping heat energy up a temperature hill much as a liquid pump forces water up a gravity hill. Such a heat pump is called a *refrigerating machine*. In its practical form, it is used to manufacture ice or to maintain the temperature of cold storage rooms at a pre-determined low value.

95. Carnot Cycle on the TN -Plane. In some respects, there are decided advantages in employing the TN -plane as an aid in

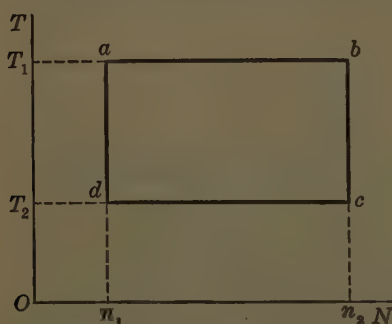


FIG. 35.—THE CARNOT CYCLE ON THE TN -PLANE

the study of heat engine cycles. In the Carnot cycle, the first and third operations are isothermal changes. On the TN -plane (Fig. 35), these are represented by straight horizontal lines. The second and fourth operations are adiabatic or constant heat changes, consequently constant entropy changes, which are represented by straight

vertical lines. Therefore the Carnot ideal cycle, using a perfect gas, is represented on the TN -plane by a rectangular area $abcd$.

The efficiency equals

$$(236) \quad \frac{\text{Heat received} - \text{Heat rejected}}{\text{Heat received}}.$$

Or

$$(237) \quad e_c = \frac{T_1(n_2 - n_1) - T_2(n_2 - n_1)}{T_1(n_2 - n_1)} = \frac{T_1 - T_2}{T_1}.$$

In some respects, this method of deriving the expression for the Carnot cycle efficiency is preferable. It is very brief and concise. Since the work area, which represents heat units and not foot-pounds, is rectangular, the use of the calculus is not required in evaluating the area. However, it is very desirable to know the relations between pressure and volume at the beginning and at the end of the various operations. This can only be done by use of formulas representing curves on the PV -plane.

96. Carnot's Principle. Equation (237) is Carnot's principle stated in symbolic language. Another statement which, perhaps, is more complete is: Whenever heat energy is transferred from a hot body to a cold body by means of some working substance which passes through a closed cycle of operations, a portion of that heat may be transformed into mechanical work while the remainder is lost to the cold body. The percentage of the energy which may be so transformed depends only upon the absolute temperatures of the hot and the cold bodies. It is independent of the nature of the working substance.

The value of the Carnot's cycle efficiency, e_c , can be equal to 1.00 only when T_2 becomes absolute zero. That is, under the condition that zero heat energy is delivered to the cold body. In practice, that is an impossible condition to realize. The practical cold-body temperature for engineers cannot be lower than that of the large streams and bodies of water on the earth. In especially favored localities along the seacoast where arctic or antarctic currents bring cold water, a temperature of 32° F. is available in the winter months, and perhaps a temperature as low as 45° F. may be available in the summer months. But there are not many such favored localities where large power plants are

desired. In equatorial regions, the temperature of the sea at the surface may be 100° F. or even higher. In any case, high ideal and actual engine efficiencies can be attained only by making the temperature of the hot body as high as possible.

In any actual engine, the four ideal reversible operations of the Carnot cycle cannot be completely realized. For instance, during the first and third ideal operations, heat energy is imagined to flow into and away from the working substance without any drop in temperature. But actually, there must be some difference in temperature between T_1 of the hot body and T_1' , the temperature of the working substance in the cylinder, during the first operation if there is to be heat flow. The same is true when heat energy is discharged to the cold body. By making the temperature drops smaller and smaller, the time for the heat to flow becomes longer and longer. The departure from the ideal in this respect may be made smaller and smaller; but the ideal can never be exactly attained. In order to carry out the second and fourth ideal operations, the cylinder and piston materials must be absolutely non-conductors of heat if the changes are to be exactly adiabatic. But again no perfect heat insulator has ever been found in nature. However, the adiabatic steps may be approximated. Thus it is seen that efficiencies of actual engines can never equal the values of the ideal Carnot engine for corresponding conditions of operation. But the ideal values may be approached more or less closely.

97. The Carnot Cycle with Vapors. When the working substance is a vapor such as steam (H_2O vapor), the PV -diagram for the Carnot cycle is as shown in Fig. 36. The isothermals are straight horizontal lines but the adiabatics are curves. The cycle is carried out in a cylinder without valves; hence the weight of substance in the cylinder remains constant. During the period of heat rejection, the vapor is condensed to a liquid and re-evaporated during the addition of heat. In detail the four operations for vapors are as follows:

1. At point a (Fig. 36), the working substance in the cylinder is in the liquid state at temperature T_1 , pressure P_1 , and volume

V_1 . Heat is received from the hot body at temperature T_1 , evaporating the liquid into vapor. This produces the volume change along the line ab at constant pressure; since for wet vapors the isothermal and isobaric lines coincide. At point b , the vapor is just dry and saturated at T_1, P_1, V_2 .

2. The cylinder is then removed from the hot body, and the non-conducting cover put on. Adiabatic expansion then proceeds along the curve bc until the temperature falls to T_2 when the pressure and volume are respectively P_2 and V_3 .

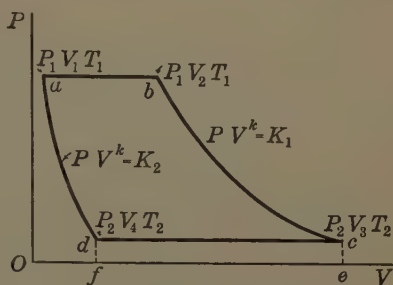


FIG. 36.—CARNOT CYCLE FOR STEAM ON THE PV -PLANE

3. The insulating cover is removed, and the cylinder is placed in thermal communication with the cold body at temperature T_2 . The vapor is compressed isothermally, and partially condensed while heat energy is rejected to the cold body. This operation is stopped at point d before condensation is complete.

4. The cylinder is then removed from contact with the cold body, and the insulating cover put on. Compression is continued adiabatically until all the vapor is reduced to the liquid state at the initial conditions as to pressure, volume, and temperature, thus closing the cycle. In order to have the cycle close exactly, as proved on page 142, it is again necessary that the following relation obtain:

$$\frac{V_2}{V_1} = \frac{V_3}{V_4}.$$

The volume of the liquid V_1 at a , Fig. 36, is so small compared to the volume of the vapor V_2 , at point b , that it must be shown by a volume scale very much out of proportion with the other volumes of the figure. If drawn to the same scale as the other volumes, V_1 would fall so nearly upon the axis of ordinates that it could not be perceived.

Carnot cycles for dry and saturated steam, for wet steam, and for superheated steam are shown upon the TN -plane in Fig. 37.

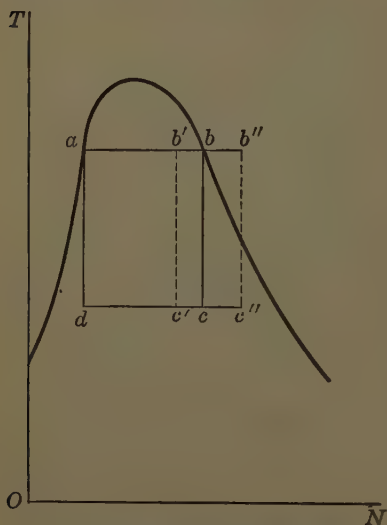


FIG. 37.—CARNOT CYCLE FOR STEAM ON THE TN -PLANE

The areas representing the cycle are rectangles in each case. The expression for the Carnot cycle efficiency is the same for all, and may be shown to be the same as for a perfect gas.

98. Clausius and Lord Kelvin.

It seems appropriate to give here a brief biographical sketch of these two men who were very largely responsible for developing thermodynamics into a true science. Rudolph Julius Emmanuel Clausius (1822–1888), a German physicist and mathematician, is famous, chiefly, for the masterly way in which

he was able to find exact mathematical expressions dealing with natural phenomena; especially those dealing with heat energy and its transformations. By restating Carnot's principle and other allied work he became largely responsible for putting the dynamical theory of heat into a form which could be generally accepted by scientists. James Clerk Maxwell called him the founder of the kinetic theory of gases. Clausius was the first to determine, mathematically, the length of the mean free path of a molecule of gas. He also did much fundamental mathematical work in the field of electrolysis. He filled most creditably the chair of physics at the University of Zürich and at Bonn. His writings are characterized by their simplicity of form and depth of thought.

William Thomson, Lord Kelvin¹ (1824–1907), was without doubt, the greatest English² physicist, and one of the greatest of all time. He received his early education from his father, who was a professor of mathematics at Glasgow University. When but ten years old, he

¹ See Silvanus P. Thompson, *Life of Lord Kelvin*.

² Here the word English is used in the generic sense referring to the English-speaking peoples.

entered Glasgow University. When fifteen years old, he was approved for the A.B. degree, but he did not accept it because he wished to enter Cambridge as an undergraduate. He received his degree of A.B. from Cambridge with very high honors at the age of twenty-one. The next year, but before he had completed the work for the Master's degree, he was elected to the chair of Physics at Glasgow University. He filled this chair for 53 years with great distinction. He brought great renown to the University, and earned many honors for himself. In 1851, he presented a paper *On the dynamical theory of heat* before the Royal Society of Edinburgh, in which the work of Rumford, Davy, Carnot, Joule, and Mayer was harmonized. In this paper, he formulated his statement of the second law of thermodynamics. His many contributions to the science of thermodynamics are his most important work. But he was one of the most versatile and prolific of scientists, and left indelible records in many departments. He possessed in the highest degree the rare combination of profound theoretical knowledge with the power of deep mathematical analysis, and the ability to invent and execute important experimental investigations.

He was the recipient of 22 honorary degrees from universities throughout the world. He was an active or honorary member in 88 scientific societies, in many of which he had held the presidency.

§9. Second Law of Thermodynamics. The second law was first stated by Rudolph Clausius¹ in 1850, and in 1851 Professor William Thomson² made a statement of this fundamental axiom, the work of Clausius on this subject being unknown to Thomson at that time.

Clausius' statement is, "It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature."

Thomson's statement of the law is, "It is impossible, by means of inanimate material agency, to derive mechanical effect from any portion of matter by cooling it below the temperature of the coldest of the surrounding objects."

A more readily understood statement of this axiom is that

¹ See Poggendorff's *Annalen*, vol. 79, 1850; and R. Clausius, *Mechanical Theory of Heat*, translated by W. R. Browne.

² *On the Dynamical Theory of Heat*, Proceedings of the Royal Society of Edinburgh, March, 1851; and Transactions of the Royal Society of Edinburgh, 1853.

heat energy naturally tends to flow downhill (temperature hill). Heat will not flow uphill unless energy in some form is expended upon it forcing it up the hill, in a way analogous to the pumping of water up a gravity hill.

This law is an axiom which cannot be proven mathematically, but experimentally it has never been found to be in error. No result based upon the acceptance of this law has ever been proven incorrect. Hence this law is justified by general observations upon the phenomena of nature, and the results produced by its acceptance.¹

100. **Reversible and Irreversible Engines.** Employing the second law as an axiom, it is now possible to prove that the

Carnot reversible engine is as efficient or more efficient than any irreversible engine. Assume the following items as shown in Fig. 38.

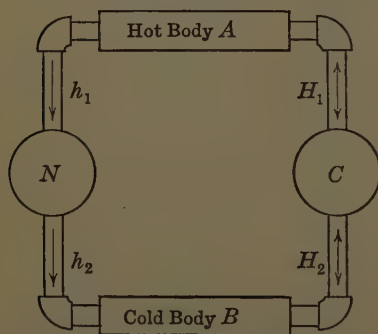


FIG. 38.—REVERSIBLE AND IRREVERSIBLE ENGINES

1. A hot body, A, which remains at temperature T_1 regardless of the amount of heat extracted.
2. A cold body, B, which remains at temperature T_2 no matter how much

heat energy may be imparted to it.

3. A Carnot reversible engine, C.
4. Any irreversible engine, N.

When operated on the direct cycle, let the following symbols represent the quantities of heat energy involved per cycle:

H_1 = B.t.u. taken from A by engine C,

H_2 = B.t.u. rejected to B by engine C,

h_1 = B.t.u. taken from A by engine N,

¹ Another statement of this law is, *In Nature entropy continually increases.* By isolating parts of the world and assuming idealized conditions, the increase may be temporarily arrested but it cannot be turned into a decrease. A. S. Eddington considers that the second law of thermodynamics holds the supreme position among all the laws of Nature. See *The Nature of the Physical World*, 1929, p. 74.

$h_2 =$ B.t.u. rejected to B by engine N ,
 e_c and $W_c =$ respectively, the efficiency and work done by C ,
 e_n and $W_n =$ respectively the efficiency and work done by N .

Then

$$(238) \quad e_c H_1 = A W_c,$$

$$(239) \quad e_n h_1 = A W_n.$$

Now assume that the work done per cycle is the same for both engines, and that engine N is more efficient than engine C . Under these conditions, engine N must take less heat from the hot body and reject less heat to the cold body than engine C . Or, if

$$(240) \quad e_n > e_c,$$

then

$$(241) \quad h_1 < H_1,$$

and

$$(242) \quad h_2 < H_2.$$

Now assume that the engines are directly connected on the same shaft and that the Carnot reversible engine is driven in reversed direction by engine N . Engine C will now take the quantity of heat H_2 from the cold body and deliver to the hot body the heat energy H_1 which it absorbed from the hot body on the direct cycle. Therefore the combined system, $N - C$, during each cycle will take from the hot body h_1 B.t.u. and return to the hot body H_1 B.t.u. The system will also take from the cold body H_2 B.t.u. and deliver to the cold body h_2 B.t.u. Referring to (241) and (242), it is apparent that this self-contained machine, *unaided by external agency*, is transferring heat from a cold body to a hot body at higher temperature. But this is contrary to the second law of thermodynamics. Therefore, by reduction to absurdity, it is proven that the irreversible engine N cannot be more efficient than the Carnot reversible engine C . Since these two engines may be any engine of the type under consideration, the conclusion that any reversible engine is as efficient or more efficient than any irreversible engine is justified.

This form of proof was used by Carnot in his original paper, but he based it upon the axiom that perpetual motion of the first class is impossible. The proof, based upon the second law, was first developed by Clausius and Thomson (Lord Kelvin).

In the same manner, it may be proved that no reversible engine can be more efficient than the Carnot. Hence the efficiency of the ideal Carnot engine is a standard of perfection for which to strive. Any actual engine which may be built can never exceed, or even equal in efficiency, the ideal Carnot engine if the second law of thermodynamics is valid.

101. Thermodynamic Temperature Scale. Up to this point in the development of the science of thermodynamics, the measurement of temperature has been based arbitrarily upon the expansion of substances such as mercury, alcohol, nitrogen, or hydrogen. But there is no substance so far found in nature which expands at a perfectly uniform rate throughout the terrestrial temperature range, say from 0° F. absolute to 4000° F. For a long time before Kelvin's day, the hydrogen gas thermometer was accepted as being the most nearly perfect one available. There were two styles of it in use, the constant-pressure and the constant-volume. In the first, the gas is confined at constant pressure, and the change in volume is used as an indication of temperature changes. For a constant-pressure Fahrenheit thermometer,

$$(243) \quad t_p = 32 + \frac{180}{v_{212} - v_{32}} (v - v_{32}),$$

in which

t_p = any Fahrenheit temperature by the constant-pressure gas thermometer,

v_{32} = the volume of the gas at the ice point,

v_{212} = the volume at the boiling point of water,

v = the volume at any temperature t_p .

The type and magnitude of the errors of the constant-pressure thermometer depends upon the gas pressure, the initial volume of gas, and upon the gas employed.

The equation for the constant-volume type of thermometer is

$$(244) \quad t_v = 32 + \frac{180}{(p_{212} - p_{32})} (p - p_{32}),$$

in which the symbols are analogous to those used in equation (243). In this type of thermometer, there is another series of errors whose magnitudes depend upon the volume of gas and the initial pressure used as well as upon the kind of gas. These many difficulties led to a search for a perfect scale which would be absolutely uniform throughout, and independent of any substance. In 1848, Lord Kelvin proposed such a scale based upon the ideal Carnot cycle. It is known as the *thermodynamic scale* or the *Kelvin scale* of temperature.

Consider the series of Carnot cycles for a perfect gas included between the two adiabatics, ad and eh ; and the two isothermals, ae and dh in Fig. 39. The discussion is much simpler if the TN -plane is used. The following conditions are assumed to be met:

(1) The first Carnot engine receives Q_1 heat units from the hot body at temperature T_1 and discharges Q_2 heat

units at T_2 to the next engine. The second engine discharges Q_3 heat units at temperature T_3 and so on for each engine in the series. (2) The quantities of heat converted into mechanical work by each engine are all equal. Hence

$$(245) \quad \begin{aligned} Q_1 - Q_2 &= q_1, \\ Q_2 - Q_3 &= q_2, \\ Q_3 - Q_4 &= q_3, \end{aligned}$$

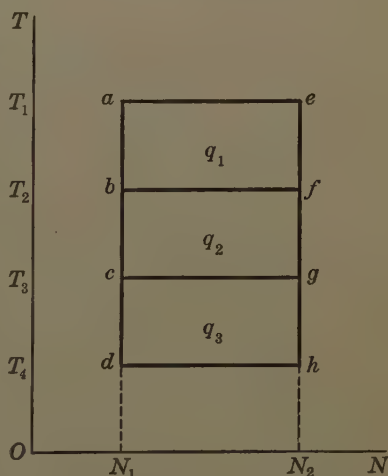


FIG. 39.—THERMODYNAMIC TEMPERATURE SCALE

and so on to the end of the series. In this series it is also specified

that

$$(246) \quad q_1 = q_2 = q_3 = \dots$$

But referring to Fig. 39 it is seen that

$$(247) \quad \begin{cases} q_1 = (T_1 - T_2)(N_2 - N_1), \\ q_2 = (T_2 - T_3)(N_2 - N_1), \\ q_3 = (T_3 - T_4)(N_2 - N_1). \end{cases}$$

From (246) and (247) the following equation results:

$$(248) \quad (T_1 - T_2) = (T_2 - T_3) = (T_3 - T_4) = \dots$$

That is, all the temperature intervals are exactly equal. (3) The last engine in the series is assumed to discharge no heat to a cold body whose temperature is absolute zero; that is, it utilizes all the heat supplied to it and it thus operates with an efficiency of 100 per cent. Any lower temperature for the cold body would result, in this case, in an efficiency greater than 100 per cent, which is absurd. Therefore a temperature zero as defined above must be the absolute zero. (4) The series of Carnot cycles cannot be extended downward indefinitely because each engine transforms a portion of the original, limited supply of heat Q_1 into mechanical work. There can then be only a number of steps equal to Q/q . At the end of this series, the discharge temperature of the last engine will be absolute zero. If the temperature of the hot body be that of melting H_2O ice and the number of Carnot cycles down to absolute zero temperature be equal to

$$(249) \quad \frac{Q}{q} = 491.6,$$

then the temperature intervals between the successive isothermals would be equal to one degree on the absolute Fahrenheit scale. If for this interval

$$(250) \quad \frac{Q}{q} = 273.13,$$

the thermodynamic Centigrade scale is defined. The series of Carnot cycles may be extended indefinitely above the melting point of ice and all temperature intervals will be exactly equal so

long as the successive q 's have the same value. It should be noted that temperatures on the thermodynamic scale are always measured above the absolute zero. The thermodynamic scale of temperature corresponds exactly with the absolute scale of the gas thermometer for an ideally perfect gas. The constant-volume hydrogen scale between 32° F. and 212° F. agrees very closely with the thermodynamic or Kelvin scale. But the Kelvin scale possesses the advantages that it is independent of the properties of any particular substance, and temperatures are expressed in terms of energy. It is also independent of any hypothesis concerning the nature of heat energy.

102. The Rankine Ideal Cycle. No ideal cycle can possibly be more efficient than the Carnot, but it is not a practical cycle upon which to operate a steam engine or turbine. The principal difficulty is that the quantity of work which can be developed per cycle is extremely small unless the engine cylinder is made very large. However, as a standard of perfection at which to aim on the basis of efficiency alone, no engine can be better than the Carnot. But it must be remembered that under the normal conditions prevailing on the earth an efficiency of 100 per cent cannot be realized even in the ideal Carnot engine.

Rankine proposed the ideal cycle for the steam engine which is pictured in Fig. 40. It is a modified Carnot-cycle engine. This cycle possesses the distinct advantage that from 10 to 20 times more power can be transformed per cycle than in a Carnot engine having the same cylinder dimensions. This is accomplished with only a slight reduction of efficiency. The Rankine-cycle efficiency is now generally used as the standard of perfection with which the performance of engines and turbines are compared.

For greater convenience, it is assumed that all four operations of the Rankine cycle occur within a closed cylinder; that is, the weight of the fluid in the cylinder is constant. It is also assumed that the cycle is performed with one pound of the substance. During certain instants in the cycle, the fluid (H_2O) is all in the vapor state, at another stage it is entirely liquid, while at other stages it is a mixture of liquid and vapor in varying proportions.

First Operation. Starting at point a , Fig. 40, when all the H_2O is in the liquid state at the boiling point, T_1 absolute temperature, heat energy is added at constant pressure until the liquid is just completely evaporated and external work is done by the forward movement of the piston. It happens that for wet or saturated vapors the isopiestic and isothermal paths, on the PV -plane, coincide. Hence the path ab represents the history of this change. The end point of this step in the cycle would be indicated by b' , Fig. 40*b*, if the operation were stopped

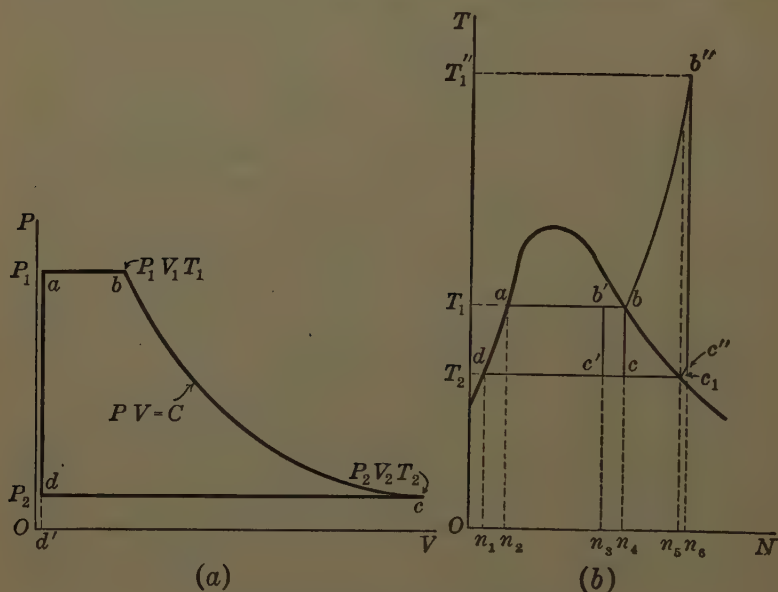


FIG. 40.—THE RANKINE CYCLE

before the liquid had been completely evaporated. The end would be represented on the TN -plane by the point b'' if the steam became superheated. The superheating stage does not proceed at constant temperature, but at constant pressure, hence this part of the path is represented by the logarithmic curve bb'' . In this last phase the Rankine cycle departs from the Carnot.

Second Operation. The heat supply is cut off at b' , b , or b'' as the case may be, and expansion proceeds along the adiabatic $b'c'$, bc , or $b''c''$. If the expansion is along $b''c''$, the steam is still

superheated at c'' , and from c'' to c_1 both the temperature and entropy decrease.

Third Operation. At c' , c , or c'' , the cylinder is instantly placed in thermal communication with the cold body, and heat is discharged by compression at constant pressure until the vapor is entirely reduced to the liquid state. This operation is represented by the line $c'd$, cd , or $c''c_1d$, depending upon whether the steam is wet, dry and saturated, or superheated at the beginning of the third operation. This operation differs from the corresponding Carnot operation in two respects. It is always continued until the vapor is completely liquefied, and if the steam is still superheated at the end of the second operation the compression which comprises the third operation is not isothermal, but it is isopiestic. However, superheat at the beginning of the third operation is undesirable and is rarely met with in practice; hence this compression step is generally isothermal as well as isopiestic.

Fourth Operation. The liquid is now compressed at nearly constant volume from pressure P_2 to pressure P_1 , and from temperature T_2 to temperature T_1 along the line da , thus closing the cycle. In the actual cycle of the steam engine, this step corresponds to pumping the water from the condenser back into the boiler, and raising its temperature from that of the exhaust to that in the boiler. The work of compression during this step is so small that it is generally considered to be negligible. On the PV -diagram of Fig. 40a, the distance od' represents the volume (not to scale in this instance, however) of one pound of water. The volume of one pound of dry and saturated steam at any pressure is so much larger in comparison that the PV -diagram is generally drawn as though the volume of the water at point d were zero. No sensible error is introduced by so doing.

↙ **103. Efficiency of Rankine Ideal Cycle.** The efficiency of the Rankine ideal cycle, using initially dry and saturated steam, may most conveniently be determined as follows (refer to Fig. 40b): As in the case of the Carnot cycle, the efficiency is

$$(251) \quad e = \frac{\text{Result}}{\text{Effort}}.$$

Then

$$(252) \quad e_R = \frac{\text{Heat received} - \text{Heat rejected}}{\text{Heat received}} = \frac{H_1 - H_2}{H_1}.$$

In equation (252),

$$(253) \quad H_1 = r_1 + h_1 - h_2,$$

and

$$(254) \quad H_2 = x_2 r_2.$$

In equations (253) and (254), the symbols represent the following quantities for one pound of steam:

r_1 = latent heat of evaporation of steam at the initial pressure,

h_1 = the heat of the liquid above 32° F. at the initial pressure,

r_2 and h_2 = the corresponding quantities at the final or exhaust pressure,

x_2 = the quality at the beginning of the third operation as at point c .

Then

$$(255) \quad e_R = \frac{r_1 + h_1 - h_2 - x_2 r_2}{r_1 + h_1 - h_2}.$$

In any practical steam engine, it is not possible to determine the value of x_2 at point c , Fig. 40a, but the TN -diagram, Fig. 40b, is much more convenient to use in computing the heat rejected. It should be remembered that areas on the TN -plane represent heat quantities in B.t.u. Therefore, in Fig. 40b,

$$\text{area } n_1 d a b n_4 = r_1 + h_1 - h_2,$$

and

$$\text{area } n_1 d c n_4 = x_2 r_2 = T_2(n_4 - n_1).$$

In the last expression, n_1 and n_4 equal the respective values of entropy at the points d and c , and T_2 is the absolute temperature during isopiestic compression. Then equation (255) becomes

$$(256) \quad e_R = \frac{r_1 + h_1 - h_2 - T_2(n_4 - n_1)}{r_1 + h_1 - h_2}.$$

In case the steam supplied is wet at the end of the first opera-

tion at b (Fig. 40a) which corresponds to the point b' (Fig. 40b), the Rankine ideal cycle efficiency equals

$$(257) \quad e'_R = \frac{\text{area } n_1dab'n_3 - \text{area } n_1dc'n_3}{\text{Area } n_1dab'n_3}.$$

And in case the steam is superheated at the end of the first operation, the ideal efficiency is the ratio of the corresponding areas for that condition.

At various times, many other cycles for heat engines have been purposed by different scientists. Stirling, Joule, Ericsson, and Lorenz are among those who proposed such cycles. But most of the cycles proposed have been proved impractical and are now of interest only as special thermodynamic problems, and because of their historical associations. Their discussion is outside the scope of this book.¹ Under the chapter on internal combustion engines two other cycles, the Otto and the Diesel, will be discussed.

104. Illustrative Problem. Compute the Carnot and Rankine ideal-cycle efficiencies for the following conditions. The engines are supplied with dry and saturated steam at an initial pressure $p_1 = 135$ lbs. per sq. in. gage; and it is exhausted at $p_2 = 2$ lbs. per sq. in. abs. The barometer reading is 29.80 in. of Hg. Then $p_1 = 135 + (29.80 \times 0.491) = 149.63$ lbs. per sq. in. abs. In this case it will be sufficiently accurate to call it 150.0 lbs. per sq. in. abs.

By referring to steam tables, the boiling-point temperatures corresponding to the absolute pressures p_1 and p_2 respectively are found to be $t_1 = 358.4^\circ \text{ F.}$ and $t_2 = 126.1^\circ \text{ F.}$ Hence

$$T_1 = 358.4 + 459.6 = 818.0^\circ \text{ F. abs.,}$$

and

$$T_2 = 126.1 + 459.6 = 585.7^\circ \text{ F. abs.}$$

By referring to the TN -plane of Fig. 40b, it is seen that n_1 equals the entropy of the liquid at absolute temperature T_2 ; and n_4 equals the total entropy of dry and saturated steam (entropy of the liquid plus the entropy of evaporation) at absolute pressure p_1 and absolute temperature T_1 . Again referring to the steam table, it is seen that $n_1 = 0.1750$; $n_4 = 1.5690$; $r_1 + h_1 = 863.1 + 330.4 = 1193.5$; and

¹ For a discussion of these cycles, see W. D. Ennis, *Applied Thermodynamics*; R. H. Thurston, *A Manual of the Steam Engine*.

$h_2 = 94.0$. Therefore

$$e_c = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{T_1} = \frac{358.4 - 126.1}{818.0} = 0.2842 = 28.42 \text{ per cent;}$$

and

$$e_R = \frac{r_1 + h_1 - h_2 - T_2(n_1 - n_1)}{r_1 - h_1 - h_2} = \frac{1193.5 - 94 - 585.7(1.5690 - .1750)}{1193.5 - 94} = 0.2590 = 25.90 \text{ per cent.}$$

The above results are typical for the two ideal cycles. The Rankine-cycle efficiency is always somewhat smaller than the Carnot-cycle efficiency.

PROBLEMS

1. Prove that no reversible-engine cycle can be more efficient than the Carnot.

2. Draw a diagram on the PV -plane representing the Rankine cycle with superheated steam.

3. Compute the Carnot and Rankine ideal-cycle efficiencies for dry and saturated steam under the following conditions: $p_1 = 250$ lbs. per sq. in. gage; exhaust is into a condenser where the vacuum is 28 in. of mercury with the barometer reading 29.50 in. of mercury.

4. Repeat Problem 3, all the conditions being the same except that steam is initially superheated 150°F . Draw a representative PV -diagram.

5. Repeat Problem 3 for a quality of 0.95, all other conditions remaining the same.

6. A steam engine uses 22 lbs. of steam for each horsepower-hour under the following conditions: $p_1 = 140$ lbs. per sq. in. gage; $x_1 = 0.98$; $p_2 = 2$ lbs. per sq. in. abs. Assuming that the heat is supplied above the temperature of the exhaust (that is, the feed water may be pumped into the boiler at the exhaust temperature), compute: (a) Carnot and Rankine-cycle efficiencies; (b) actual thermal efficiency.

7. Compute the pounds of steam which the ideal Carnot engine would require to develop one horsepower-hour under the conditions given in Problem 6. Repeat for an engine whose efficiency is 100 per cent.

8. Referring to Fig. 34, page 141, use the following data to compute (a) Carnot-cycle efficiency; (b) the mean effective pressure of the Carnot PV -diagram when air is the working medium; $p_1 = 500$ lbs. per sq. in. abs., $V_1 = 15$ cu. ft.; $V_2 = 25$ cu. ft.; $t_1 = 1050^\circ \text{F}$; $t_2 = 50^\circ \text{F}$.

(HINT. Mean effective pressure equals the net work area divided by $V_2 - V_1$, but use proper precaution regarding units.)

CHAPTER XII

FUELS AND COMBUSTION

105. Combustible and Combustion. The vast quantities of heat energy required to produce mechanical and electrical power on the enormous scale demanded by the modern industrial developments has to be obtained by burning fuels in air or by combustion. "Combustion is a chemical process accompanied by the liberation of light and heat, commonly the union of substances with oxygen. It is the act of burning."¹ The substance which is burned is called *combustible*. Oxygen is called the *supporter of combustion*. The combustion which liberates the heat to generate power is very rapid, and temperatures produced thereby range from 1800° F. to 3000° F. or even higher. Combustion does take place, under certain conditions, at a slow rate and accompanied by only a slight temperature rise. The total B.t.u. liberated per pound of fuel is probably the same for both types of combustion. There is also a very slow combustion taking place with very slight temperature rise, as for instance when a linseed oil paint dries. But engineers are not interested in slow rates of combustion where generation of power by means of heat engines is concerned.

Many substances burn readily and rapidly in air, and much more intensely in pure oxygen, which is the only constituent of the air supporting combustion. Several other elements are supporters of combustion also. The union of hydrogen (a combustible) and chlorine (a supporter of combustion) is a combustion process producing hydrochloric acid accompanied by the liberation of heat energy. Practically every chemical reaction is accompanied by liberation or absorption of heat energy. But only those reactions which employ oxygen as a supporter of combustion, proceed with great rapidity, and liberate large quantities of heat per unit of weight at high temperatures

¹ From Webster's *New International Dictionary*.

are of interest to the engineer. In general, those chemical changes which are accompanied by liberation of heat energy are called *exothermic* reactions. Those changes which take up heat energy from the surroundings are called *endothermic* reactions. Every combustion is an exothermic change but many exothermic changes are not combustions in the estimation of engineers.

In two or three instances, Nature has been very liberal in furnishing a store of combustible material and the supporter of combustion, both of which are readily accessible to man. The important element in all fuels is carbon. An abundant supply of oxygen is available in the atmosphere to be had merely by reaching out and taking it. If by an accident of Nature oxygen had been one of the rare gases hard to isolate and available in small quantities only, modern central power stations would never have been developed along the present lines. It is indeed a fortunate circumstance that a plentiful supply of carbon is available at low cost, and all the oxygen needed may be had free of all first cost.

Solid and liquid fuels as found in nature are not pure carbon but contain from 5 to 15 per cent hydrogen as well as earthy matter which forms the ash content. Sulphur in small amounts is generally present also. However, it is considered an undesirable impurity, since it burns to sulphur dioxide, which in turn combines with the water vapor of combustion to form sulphurous or sulphuric acid. These acids corrode iron and steel rapidly. The hydrogen is combined chemically with the carbon to form the hydrocarbon compounds. Solid fuels contain moisture held mechanically within the pores and combined in the form of water of crystallization. The value of any fuel in power transformations depends upon the number of B.t.u. which are liberated when one pound of the fuel is burned in a furnace, or in the cylinder of an internal combustion engine. The quantities of impurities in the form of moisture, ash, and sulphur carried by the fuel are factors of secondary importance in establishing the value of the fuel. The heating values of several fuels which have a definite chemical formula are given in Table 17. They are the English equivalents of the values given in Julius Thomsen's *Thermochemical Investigations*.

TABLE 17
HEAT OF COMBUSTION OF VARIOUS FUELS¹

COMBUSTIBLE	HIGHER HEATING VALUE, B.T.U.	
	Per Pound	Per Cu. Ft.
Carbon to CO ₂	14 545	
Carbon to CO	4 350	
Carbon monoxide to CO ₂	4 370	324
Hydrogen to H ₂ O	62 000	328
Methane, (CH ₄), to CO ₂ and H ₂ O	23 840	1 010
Ethane, (C ₂ H ₆), to CO ₂ and H ₂ O	22 230	1 765
Hexane, (C ₆ H ₁₄), to CO ₂ and H ₂ O	20 920	4 760
Benzene, (C ₆ H ₆), to CO ₂ and H ₂ O	18 450	3 809
Methyl alcohol (CH ₄ O)	10 250	873
Ethyl alcohol (C ₂ H ₅ O)	13 325	1 622
Sulphur to SO ₂	4 000	

106. **Fuel Classifications.** Fuels may be divided, in a general way, into three classes, depending upon their natural state; namely, the *solid, liquid, and gaseous*. Each class of natural raw fuels may be burned as it is mined; or the raw fuels may be processed or manufactured with an increase in the efficiency of utilization besides recovering by-products sufficiently valuable in most cases to pay for the processing. In a few instances, the

TABLE 18
FUELS

SOLID	LIQUID	GASEOUS
Bagasse	Alcohols	Manufactured gas
Briquette (fuel)	Oils, animal	Natural gas
Charcoal	Oils, vegetable	
Coal	Petroleum	
Coke	Petroleum products	
Lignite		
Sawmill refuse		
Straw		
Tan-bark		
Wood		

¹ The volumes of gases and vapors are for 60° F. and a pressure of 30 inches of mercury. The products of combustion are reduced to 60° F.

fuel itself is the by-product of a manufacturing process, as in the cases of tan bark, sawdust, and bagasse (sugar cane from which the juice has been extracted).

Table 18 contains the names of the more important fuels in each class.

107. Fuels of Minor Importance. Straw, wood, spent tan-bark, sawmill refuse, bagasse, and the like are fuels which are of importance in certain localities and industries. In the aggregate they constitute only a small percentage of the total fuel used for generating power. Their fuel value is due to their cellulose content. Cellulose is a carbohydrate having the chemical formula $(C_6H_{10}O_5)_x$. This is the same percentage composition as starch. These fuels generally carry large percentages of moisture due to their cellular structure. *The heating value of any fuel decreases in direct ratio to the increase in percentage of moisture content.* The heating values of one pound of the various fuels are given in Table 19.

Wood is a vegetable fiber which has undergone no changes due to decay or distillation. Woods are divided into two broad classes, hard woods and soft woods. The former class comprises ash, beech, elm, gum, hickory, maple, and oak. The soft woods include cedar, fir, pine, and spruce. Due to the presence of oils, gums, and resins, the soft woods generally give the higher heating values per pound of dry wood. At the present time sawmill refuse (sawdust, slabs, and hogged wood) is the principal source of this kind of fuel.

Boiler furnaces of special design are necessary in order to burn successfully fuels obtained directly from vegetable growth. The chief factor involved in the rapid combustion of these fuels on a large scale is large furnace volume. Two or three times the volume which is satisfactory for coal or oil is necessary.

Bagasse. To save the cost of handling, this fuel must be burned directly from the rolls of the cane mills. In this condition it is very wet, containing in some cases as high as fifty per cent of moisture. Green bagasse produces the best results when burned in a very large furnace and on a hearth. It also requires air-blast at pressures higher than can be readily produced naturally by a stack. Preferably, the air for combustion should be preheated. But a fairly satisfactory furnace with hearth has been developed which permits the use of cold air.¹ The effective B.t.u. per pound of wet bagasse for steam

¹ See the Babcock and Wilcox Company's book, 36th edition, *Steam*, p. 243.

generation usually varies from 3500 to 4500. The difference between the heating value per pound of dry bagasse as given in Table 19 and the above-mentioned values is required to evaporate and superheat the moisture which it contains.

Tan-Bark. This fuel is the fibrous residue after the bark has been leached in leather manufacture. Wet tan-bark contains from 60 to 70 per cent of moisture and its effective heating value is from 2500 to 3000 B.t.u. per pound of wet bark. This fuel must be burned in large furnaces usually provided with a Dutch-oven effect. This gives a large area of heated brickwork radiating heat to the freshly fired fuel in order to dry it out and ignite it.

Straw. On the large grain ranches of the Western prairies where other fuels are very expensive, straw from which the grain has been threshed is used to some extent for agricultural power.

Charcoal. Charcoal is produced from wood by destructive distillation in an insufficient air supply for complete combustion. The distillation was originally accomplished by arranging the wood in a conical pile or mound, covering it with earth, and igniting it. Just sufficient air is supplied to burn the volatile products driven off from the wood after the pile is once kindled. This furnishes about the right amount of heat to continue the distillation until all the volatiles are driven out and only the carbon residue is left. Retorts have been developed which consume much less of the volatiles, thus leaving them to be refined and utilized for other purposes. Only about 30 per cent of the original weight of wood remains in the form of charcoal. Charcoal is nearly all carbon and has a heating value of from 11000 to 14000 B.t.u. per pound. Charcoal is now more of a by-product

TABLE 19
REPRESENTATIVE ANALYSES OF FUELS OF MINOR IMPORTANCE
(Per Cents by Weight in Dry Fuel)

FUEL	CARBON	HYDRO- GEN	NITRO- GEN	OXYGEN	ASH	B.T.U. PER LB. OF DRY FUEL	% MOIS- TURE IN 1 LB. OF WET FUEL
Woods	50.0	6.0	0.10	43.5	0.4	8000-9100	15-50
Straw	43.5	6.0	0.5	45.0	5.0	5500-6500	10-15
Tan bark	51.8	6.0		40.8	1.4	9 500	50-65
Bagasse	45.0	6.0	0.30	47.2	1.5	8 400	40-50
Charcoal	70.0	3.5	0.80	13.7	13.0	12 500	5-10

since most distillation of wood is carried on in retorts in order to obtain the resulting turpentine and oils which are of greater commercial value than the charcoal.

Charcoal is seldom used for the generation of power. Its chief use is in the chemical industries to absorb gases, for which it possesses great affinity. It is also used for fuel in certain metallurgical work.

108. Coal. All forms of coal are generally supposed to be of vegetable origin. During the geologic periods designated carboniferous, at certain favored localities, there were great forest areas. Trees, ferns, and other coarse plants grew very rapidly and to enormous size. These forests were probably located in swampy regions where the fallen tree trunks, leaves, and other refuse were stored under water. This forest litter accumulated over thousands of years into great thick beds of carboniferous material. A slow decay and distillation then took place at moderate pressure and temperature continuing through long geological ages. Occasionally more or less violent disturbances of the earth's crust occurred. In some cases this decaying vegetable matter was covered to appreciable depths and was thus subjected to heavy pressure and the distillation then proceeded under heavy pressure and in some instances higher temperature. The woody material of the plants was slowly transformed into the material now called coal largely by the removal of most of the oxygen content of the wood. Reference to Table 20, page 168, will show that there is a decided loss in oxygen content and a somewhat corresponding increase in the carbon in passing from peat to anthracite coal.

The results from considerable study of the problem of the origin of coal seem to indicate that the character of the coal in different localities depends more upon the pressure and temperature to which it has been subjected during disturbances of the earth's crust than upon the time factor. Certain beds of lignite appear to have been deposited at about the same time long ages ago with certain other beds which have now become high-rank coals. The lignite beds have remained undisturbed by earth movements, while the coal beds have been greatly disturbed and buried to considerable depths, thus subjecting the material to

considerable pressure for long periods, and perhaps to moderately high temperature for a short period of time. These earth movements with the accompanying pressure and temperature seem to be the chief factors which have produced coal in one instance and lignite in the other.

109. Coal Sampling and Analysis. Analyses of all fuels are performed on a small portion called a *sample*. Great care must be exercised in selecting the sample if the results of the analysis are to be truly representative of the bulk of fuel.

Methods of sampling and analyzing coals have been extensively investigated by the United States Bureau of Mines,¹ and by Committees of the American Society for Testing Materials.² The results of long experience of those bodies of investigators in analyzing coals have been incorporated in a code of specifications. When this code is carefully followed, the analytical results for a given sample of coal as obtained by different analysts agree within very close limits. This code has been adopted as standard by nearly all chemists and engineers in the United States.

The method of sampling coal varies somewhat, depending upon the location and quantity of coal to be sampled. The sample may be taken in the mine, when it is being loaded or unloaded from the railroad car, ship, barge, or truck, or from the storage bin or pile of the dealer.

The sample is generally selected by taking small increments of equal size regularly spaced over or through the coal. Each increment varies from one to twenty pounds depending upon the quantity of coal to be sampled and its condition. The completed gross sample varies from 100 to 2000 pounds, depending upon the total tonnage sampled, the size of the coal lumps, and the amount and character of the impurities in it.

The gross sample is spread on a smooth floor, the larger lumps broken up and the sample thoroughly mixed. It is piled in a conical pile and quartered. Alternate quarters are then rejected. This process is repeated until the sample is reduced to the quantity which is to be sent to the laboratory. The final sample usually varies from 2 to 10 pounds. It is then sealed in an air- and moisture-proof container for shipment.

¹ See *Technical Papers Nos. 1, 8, and 76*, U. S. Bureau of Mines.

² See *Standards*, American Society for Testing Materials, 1924, p. 981.

110. Ultimate Analysis. There are two kinds of analysis from which information as to the value of the coal as a fuel may be obtained, namely, *ultimate chemical analysis* and *proximate analysis*. The ultimate analysis can only be made by a skilled chemist provided with rather elaborate apparatus. The elements composing the coal are accurately determined in percentages by weight together with the incombustible material called ash, and the heating value of the coal. In some cases the ash may be analyzed, especially if a troublesome clinker is produced in the furnace. The constituents generally determined by the ultimate analysis are carbon, hydrogen, nitrogen, oxygen, sulphur, and ash. Representative ultimate analyses of the different kinds of coal as taken from publications of the United States Bureau of Mines are shown in Table 20.

TABLE 20
ULTIMATE ANALYSES OF REPRESENTATIVE COALS
(Fuel Free from Moisture Taken as 100 Per Cent)

RANK	CAR- BON	HYDRO- GEN	NITRO- GEN	SUL- PHUR	OXY- GEN	ASH	B.T.U. PER LB. OF DRY COAL
Anthracite.....	87.9	2.3	0.7	0.4	1.5	7.2	13 961
Semianthracite.....	80.4	3.3	1.0	0.8	2.8	11.6	13 690
Semibituminous.....	80.5	4.7	1.3	2.1	3.4	8.0	14 373
Bituminous.....	77.2	5.0	1.5	2.0	7.2	7.1	13 882
Subbituminous.....	64.7	5.1	1.6	1.0	15.0	12.6	11 410
Lignite.....	61.8	4.3	0.9	1.1	18.3	13.6	10 500
Peat.....	21.0	8.4	1.1	0.6	62.9	6.0	8 280

111. Proximate Analysis. The proximate analysis was devised by engineers in the early days of power generation largely because ultimate chemical analyses were expensive and difficult to obtain. Besides they did not give any advance information as to how a coal would perform in the furnace under given conditions of draft, type, and size of furnace. Hence from necessity the engineer developed a method of analysis which could be performed with simple apparatus ordinarily at hand such as weighing scales and large crucibles, using rather large fuel samples. The main object was to obtain quickly an intelligent estimate of the comparative fuel values of different coals, and to predetermine in an approximate way their manner of burning in the boiler furnace. The items determined by the proximate analysis

correspond somewhat to the various stages of combustion passed through as the fuel burns in the furnace as outlined below.

When a quantity of fresh raw coal is introduced into a hot boiler furnace the following stages of combustion may be recognized: first, the coal is heated from room temperature toward the kindling temperature. The surface moisture and moisture of crystallization are driven off. The moisture probably has all left the coal by the time its temperature reaches 220° F. or 240° F. At about 250° F. the coal substance begins to vaporize in appreciable quantities. It comes off as a dense black smoke and escapes up the chimney unless suitable provision has been made in the furnace to permit combustion of this gaseous product. These vapors are called *volatile matter*. They are composed of hydrocarbons, nitrogen, and oxygen of the fuel. The volatile matter has a high heating value per pound but an adequate air supply and proper mixing of the air and fuel must take place, and the mixture must be maintained above the kindling temperature if combustion is to be complete. The volatile matter burns with a long yellow flame. After a few minutes, the volatile matter is driven off and the coke, nearly pure carbon, remains. It burns with a short hot bluish flame. This portion of the fuel minus the ash is called the fixed carbon. When the fixed carbon is completely burned, only the ash remains.

The determination, in the laboratory under rather definitely specified conditions, of moisture, volatile matter, fixed carbon, and ash expressed in percentages by weight together with the calorific value in B.t.u. of one pound of coal, constitutes the proximate analysis. For complete detailed specifications of apparatus and procedure and for the results of numerous coal analyses performed for coals from various states, consult publications of the United States Bureau of Mines.¹

The terms *volatile matter* and *fixed carbon* do not represent any particular group of compounds which exist in the coal. The method of their determination is an arbitrary one. The results for different coals or for the same coal are comparable only when the temperatures and rates of heating are the same. In a general way, the sum of the volatile matter and fixed carbon represents the combustible material in the coal. However, the oxygen and the nitrogen in the coal are largely driven off with the volatile matter. They are, of course, not

¹ U. S. Bureau of Mines, *Technical Paper No. 76; Bulletins Nos. 22, 85, 123, and 193.* Also Marks' *Mechanical Engineers' Handbook.*

true combustibles. In steam-boiler testing the word *combustible* is a technical term which denotes the sum of the volatile matter and fixed carbon as determined by the proximate analysis. This is, in a way, unfortunate but this definition has become well established by long use.

The number obtained by dividing the percentage of fixed carbon by the percentage of volatile matter is called *fuel ratio*. This term was first applied by Persifor Frazer in 1879 to the Pennsylvania coals.¹

Table 21 contains typical proximate analyses of the various ranks of coals found in the United States. The analyses shown in Tables 20 and 21 are for the same samples of coal.

TABLE 21
PROXIMATE ANALYSES OF COALS
(Wet Coal Equals 100 per Cent)

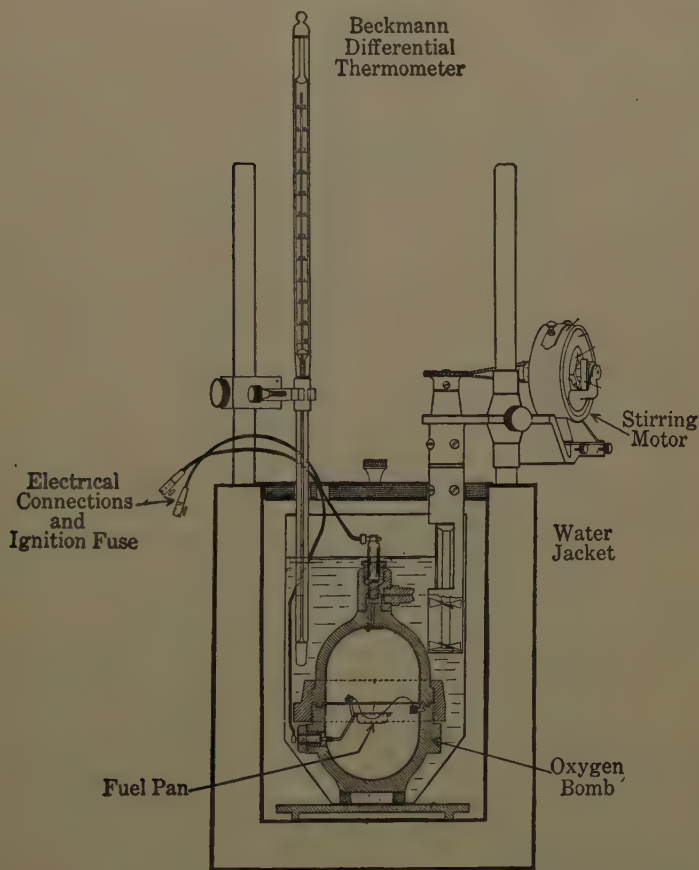
RANK	MOIS- TURE	VOLA- TILE MAT- TER	FIXED CAR- BON	ASH	B.T.U. PER LB. OF WET COAL	B.T.U. PER LB. OF COM- BUSTIBLE	FUEL RATIO
Anthracite.....	1.3	5.7	85.8	7.2	13 780	15 050	15.1
Semianthracite.....	3.5	9.3	76.0	11.2	13 215	15 480	8.17
Semibituminous.....	3.2	20.7	68.3	7.8	13 910	15 630	3.30
Bituminous.....	3.8	35.9	53.5	6.8	13 550	14 950	1.49
Subbituminous.....	20.7	33.6	35.7	10.0	9 040	13 040	1.06
Lignite.....	35.2	29.8	25.3	8.7	6 700	12 150	0.85

112. Fuel Calorimetry. The heating value of solid and liquid fuels is obtained by burning a small carefully prepared sample of the fuel in a device called a *calorimeter*. The *calorific value*, or heating value, may then be computed when two conditions are met. First, the combustion must be perfect and complete. Second, all the heat generated must be measured. The supporter of combustion may be pure oxygen, air, or some chemical compound rich in oxygen. The essential requirement is that the oxygen shall be adequate and supplied under conditions which will insure complete and perfect combustion.

The oxygen-bomb type of calorimeter, one form of which is shown in Fig. 41, gives the most accurate and reliable results. The bomb is a heavy steel shell made in two parts, and capable of withstanding

¹ "Report M M," *Second Geological Survey of Pennsylvania*, 1879, p. 144.

an internal pressure of 2000 pounds per square inch or more. The bomb is lined with a thin sheet of platinum or nickel which resists the corrosive action of acids formed during the combustion of the fuel. The volume of the bomb varies from 300 to 600 cc., depending upon the designer's ideas. A fuel pan of platinum or nickel is arranged



(Emerson Apparatus Co.)

FIG. 41.—OXYGEN-BOMB CALORIMETER

inside the bomb. The ignition of the fuel is generally accomplished by use of an electric fuse wire. The fuel sample carefully prepared and accurately weighed (usually about one gram) is placed in the pan. The fuse is then attached. The bomb is closed tightly, and filled with pure oxygen gas from a tank to a pressure of 300 or 400 pounds per square inch. The bomb is then submerged in a bucket partially

filled with a weighed quantity of distilled water. The bucket and bomb are then placed in a water jacket or a Dewar vacuum jar in order to reduce heat radiation to a minimum. The initial temperature of the water is carefully measured by a delicate thermometer indicating hundredths of a degree. Generally, temperatures are read regularly at 30-second intervals for a period of 15 minutes, 5 minutes before the charge is ignited, during the combustion period which lasts 5 or 6 minutes, and for a 5-minute post-combustion period. From these temperature readings, a correction for radiation may be readily determined. Certain other corrections for the electric fuse and formation of nitric acid must be made in case a high degree of precision is desirable in the results. During the combustion period, the water is stirred constantly by a motor-driven propeller.¹

Knowing the weights of fuel and water, and the water equivalent of the calorimeter, the temperature rise and the various corrections the calorific power of the fuel may be computed from the following heat equations:

$$(258) \quad UM_f = c(M + M_e)(t_2 - t_1 \pm r).$$

$$(259) \quad U = c \frac{(M + M_e)(t_2 - t_1 \pm r)}{M_f}.$$

In equations (258) and (259),

c = the specific heat of water (usually considered to be 1.00),

M = pounds of water in the calorimeter bucket,

M_e = water equivalent of the calorimeter in pounds,

M_f = pounds of fuel burned in the calorimeter,

r = the sum of the various temperature corrections due to electric fuse, radiation, acid formation, etc.

t_1 = initial temperature of the water and calorimeter in degrees F.

t_2 = final temperature of the water and calorimeter after combustion is complete,

U = calorific value of 1 lb. of fuel in B.t.u.

The specific heat of water is generally considered to be 1.00. Since the temperature of the water is generally in the neighborhood of 70° F. and the rise during combustion is small (from 3 to 5 degrees), the error due to this assumption is negligible.

¹ For complete details of preparing fuel sample, conducting the test, and making all computations see Journal of the American Chemical Society, vol. 25 (1903), p. 659. *Scientific Paper No. 230*, 1914, Bureau of Standards. *Circular No. 11*, 1917, Bureau of Standards. Richards and Barry, Journal of the American Chemical Society, vol. 37, p. 993.

The action of the bomb calorimeter is intermittent and not suitable for use with gaseous fuels. A continuous flow calorimeter or gas calorimeter is used to determine the heating power of gases. Views of a gas calorimeter are shown in Figs. 42 and 43.¹

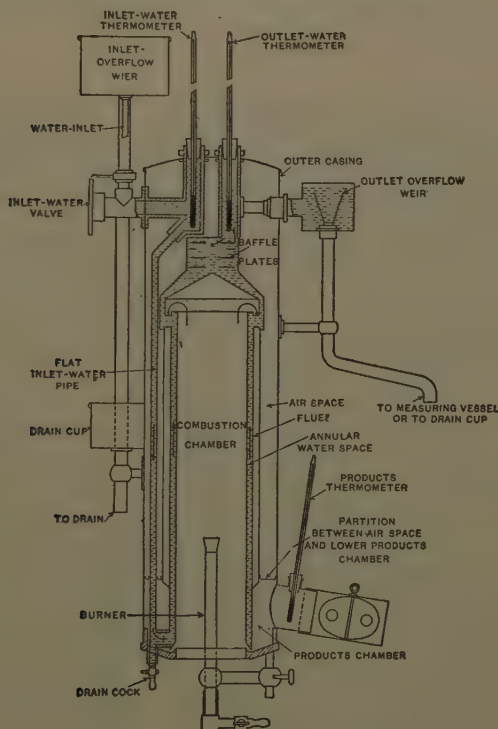


FIG. 42.—GAS CALORIMETER

Because of their molecular state, gases burn readily and completely in a properly designed burner, taking the necessary oxygen from the air. The heat of combustion is transferred from the hot gases leaving the burner to flowing water inside a series of thin copper tubes. The tubes are placed in a circular arrangement and are covered by a jacket to prevent heat loss by radiation. There is counter-flow of hot gases on the outside of the tubes and the water inside; thus in the operation of the calorimeter the gas burns continuously, and a uniform flow of water through the tubes is maintained. The flow of water and gas are so regulated as to maintain nearly constant inlet and discharge

¹ For complete details on gas calorimeters as to their construction and operation see *Circular No. 48* and *Technologic Paper No. 36*, U. S. Bureau of Standards.

temperatures of the water. This difference in temperature is usually between 5° and 15° F. From the measurements of gas volume, gas pressure, weight of water heated, and the mean temperature rise of the water flowing, the heating value of the gas in B.t.u. per cubic foot of standard gas at a temperature of 60° F. and a pressure of 30 inches of mercury may be readily computed.

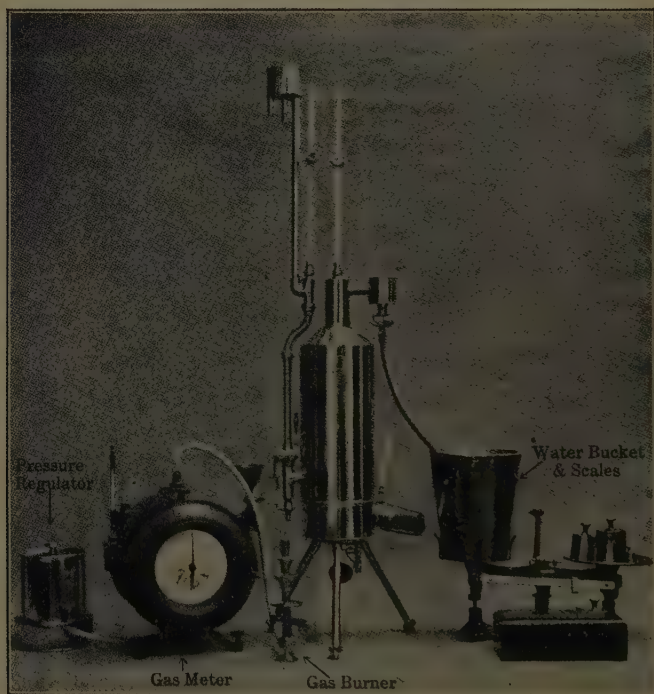


FIG. 43.—GAS-CALORIMETER SET-UP

The heating values of fuels as determined by calorimeter are ideal values. In the case of solid and most liquid fuels, this calorific value is determined by burning the fuel in an excess supply of pure oxygen at constant volume, under a considerable and variable pressure. The heating value thus obtained is undoubtedly considerably higher than could possibly be realized in any boiler or other furnace in which the combustion is carried on at nearly constant pressure with oxygen taken from the air. Again, two coals, differing widely as to analysis and calorific power as shown by the oxygen bomb calorimeter, might

conceivably give the same evaporative efficiency when burned in a boiler furnace because the design of the furnace, the type of stoker, or the conditions of operation may, in a particular plant, be better adapted to the inferior fuel. Hence the true evaporative power of fuels has to be determined by trial under the exact conditions of plant equipment and operation.

113. Higher and Lower Heating Values of Fuels. Practically all fuels contain more or less hydrogen. Coals usually contain from 3 to 6 per cent, fuel oils contain from 10 to 15 per cent, and some gaseous fuels may contain as high as 50 per cent or even more of hydrogen.

One pound of hydrogen unites with 8 pounds of oxygen to form 9 pounds of water thus,



$$(261) \quad 4 + 32 = 36 \text{ (molecular weights),}$$

Or

$$(262) \quad 1 + 8 = 9.$$

When the heating value of a fuel is determined by use of an oxygen-bomb calorimeter, all the products of combustion are cooled to approximately room temperature, and the water of combustion which may be in the vapor state when formed is all condensed. Hence it gives up its latent heat of evaporation and the heat of the liquid between the boiling point and room temperature. All this heat due to condensation of the water vapor is added in to give the total heat of combustion, or the *higher calorific value*. When fuel is burned in a furnace or in an internal combustion engine, all the products of combustion at a temperature considerably above 212° F. are discharged into the atmosphere. Hence the water formed by combustion of the hydrogen escapes in the form of superheated water vapor. Because of the low partial pressure of the vapor in the mixture of stack gases the temperature at which it would condense (that is, its boiling point at its pressure) is in the vicinity of 100° F. This corresponds to a vapor pressure of approximately one pound per square inch absolute. The heat which is carried away in this superheated vapor

in effect reduces the calorific value of the fuel as measured by oxygen-bomb calorimeter. This gives rise to the idea of a computed *lower heating value*. There has been considerable discussion and much confusion exists as to the exact temperature datum above which the heat of the escaping vapor should be computed. Many engineers, especially in Europe, contend that the lower heating value is the only true basis upon which to determine boiler and gas-engine performance because they are inherently incapable of cooling the products of combustion to, or below, the temperature of condensation through no fault of their own. American engineers have, so far, considered this loss to be due to the imperfection of the prime mover and its auxiliaries. Hence in the United States all engine and boiler efficiencies are based upon the higher heating value. The American Society of Mechanical Engineers in its *Power Test Codes* specifies the use of the higher heating value. The existence of a lower heating value is recognized, but no general agreement has been reached as to the datum above which it should be computed. However, equations (263) and (264) are suitable for computing the lower heating value.

$$(263) \quad u = 9 H[r + h - h_a + \bar{c}_p(t_s - t)],$$

and

$$(264) \quad U_1 = U - u$$

in which

U = higher heating value in B.t.u. as determined by oxygen-bomb calorimeter,

U_1 = lower heating value in B.t.u. per lb. of fuel,

u = B.t.u. loss per lb. of fuel due to uncondensed moisture,

H = lbs. of hydrogen per lb. of fuel,

r = latent heat of evaporation per 1 lb. of vapor at its temperature of evaporation,

h = heat of the liquid above 32° F. for 1 lb. of water at the temperature of evaporation,

h_a = heat of the liquid above 32° F. for 1 lb. of water at atmospheric temperature,

t = temperature of evaporation of the vapor at its partial pressure in the products of combustion, degrees F.,

t_a = temperature of the atmosphere, degrees F.,

t_s = the temperature, degrees F., of the products of combustion leaving the boiler or engine,

$\bar{c}_p$ = the mean specific heat of the vapor from t to t_s .

If average values of t_a and t are chosen as 75° F. and 100° F. respectively, (263) becomes

$$(265) \quad u = 9 H[1060 - 0.46(t_s - t)].^1$$

The values of t_a and t may vary 20 degrees F. above or below those chosen without seriously affecting the value of u in equation (265). The lower heating value of coals under average conditions of operation prevailing in modern central power stations varies from 95 to 97 per cent of the higher value. For fuel oil, it is about 93 to 94 per cent of the higher value. In the case of gas, manufactured or natural, the lower value may become as small as 85 per cent of the higher depending upon the gas composition and the conditions of plant operation.

114. Heating Values Computed from Analyses of Fuels. Many formulas, by means of which the heating values of fuels may be computed from the results of ultimate or proximate analyses, have been proposed from time to time. Probably the most used formula is known as Dulong's formula. This formula was probably proposed by Berthelot or Mahler² in memory of Pierre Louis Dulong (1785-1838), French chemist and physicist. Dulong did some work, the results of which were published in 1838 after his death.³

Dulong's formula is founded on the principle that each combustible element in the fuel generates the same quantity of heat that it would if it were burned alone in a pure state in an oxygen-bomb calorimeter. The formula gives results for composite fuels which usually agree with those of the calorimeter within 2 or 3 per cent. The formula as adopted in the *Power Test Code* of the American Society of Mechanical

¹ See Mark's *Mechanical Engineers' Handbook*, p. 337, for the value of $\bar{c}_p$. G. F. Gebhardt, *Steam Power-Plant Engineering*, p. 95. *Bulletin 223*, 1923, U. S. Bureau of Mines, pp. 46-47.

² P. Mahler, *Contribution à l'Etude des Combustibles*, a bulletin of the Société d'Encouragement pour l'Industrie Nationale. Paris, 1892.

³ Pierre Berthelot, *Traité Pratique de Calorimétrie Chimique*. Académie des Sciences, Comptes Rendus, vol. 7, 1838, p. 871.

Engineers is as follows.¹

$$(266) \quad U = 14600C + 62000 (H - O/8) + 4000S.$$

In (266), the symbols C , H , O , and S refer respectively to the decimal fraction of a pound of carbon, hydrogen, oxygen, and sulphur in one pound of fuel.

Dulong's formula is based upon the assumption that all the oxygen of the fuel is united with enough of the hydrogen to form water. This assumption is not fulfilled in some fuels such as wood, peat, certain cannel coals, and fuels high in oxygen. For these fuels the formula may give results which are in error as much as 14 per cent.²

Many attempts have been made by investigators to devise more or less empirical equations by use of which the calorific value of coals might be computed from the proximate analysis. But no satisfactory equation of universal application has been found. William Kent³ proposed a series of values based upon combustibles which are fairly satisfactory for the coals of eastern United States. These values may be plotted to give Kent's curve.

Goutal's formula⁴ as modified by De Paepe⁵ has been much quoted, and much used. It is probably as accurate as any formula, using proximate analysis results, can be. Goutal's formula transformed to give B.t.u. per pound of dry coal is

$$(267) \quad U = \frac{14670 C + AM}{100}.$$

De Paepe's modification of Goutal's formula in English units is

$$(268) \quad U = \frac{14670 C + A'M'}{100}.$$

The symbols represent the following quantities:

A and A' = experimental coefficients depending upon values of M and M' (Tables 22 and 23),

C = the percentage of fixed carbon in the dry coal by proximate analysis,

¹ The U. S. Bureau of Mines' publications contain the formula

$$U = 14544C + 62028 (H - O/8) + 4050S.$$

² See Wm. Kent, *Steam Boiler Economy*, 2nd ed., pp. 77, 141-146.

³ Kent's *Mechanical Engineers' Handbook*, 10th ed., pp. 357 and 362.

⁴ Académie de Sciences, *Comptes Rendus*, vol. 135, 1902, p. 477.

⁵ D. De Paepe, in *Bulletin, Association Belge des Chimistes*.

M = the percentage of volatile matter in the dry coal by proximate analysis,

$$M' = \frac{100 M}{M + C},$$

U = calorific value of one pound of dry fuel (higher value) in B.t.u.

TABLE 22

GOUTAL COEFFICIENTS

M	A
2 to 15%	23 400
15 to 30	18 000
30 to 35	17 100
35 to 40	16 200

TABLE 23

DE PAEPE COEFFICIENTS

M'	A'	M'	A'
2 to 12%	25 200	30 to 35%	16 920
12 to 17	21 600	35 to 38	14 440
17 to 24	19 710	38 to 40	14 220
24 to 30	18 380	40 to 50	13 700

De Paepe's modification of Goutal's formula generally gives results within 6 or 8 per cent of those by calorimeter for the fuels of the United States. However, formulas of this type are now used very little since there are several very good calorimeters available on the market at a moderate cost, and it is not extremely difficult to carry out a calorimeter determination with commercial accuracy, which is generally considered to permit an error of ± 2 per cent.

115. Classification of Coals.¹ The United States Geological Survey has adopted the term *rank* for the various groups into which coals may be divided on the basis of physical and chemical properties. The high-rank coals contain high percentages of carbon and the percentages of volatile matter and oxygen are low. The low-rank fuels contain large percentages of oxygen, moisture, and volatile matter; and the heating value of low-rank coals is generally low. The term *grade* is used to distinguish the amount of inorganic impurities, as ash and sulphur, in coals. A high-grade coal of any rank is one which has a low ash content.

One of the earliest classifications of coal was prepared in 1858 by Professor Henry D. Rogers, Pennsylvania State Geologist.² He found that the coals of that state could readily be divided

¹ See *Professional Paper 100-A*, United States Geological Survey, 1922, p. 309.

² *The Geology of Pennsylvania*, Vol. 2, Philadelphia, 1858.

into four ranks, namely, anthracite, semianthracite, semibituminous, and bituminous.

For many years these four groups served very satisfactorily for scientific and commercial work. But when the governmental bureaus began a comprehensive study of the coal fields of the United States about 1904, it was found that two new ranks were desirable in order to properly classify the Western coals.¹ The classification of coals generally employed in the United States now includes the following ranks:

Anthracite
Semianthracite
Semibituminous
Bituminous
Subbituminous
Lignite

The various ranks are distinguished largely by the *fuel ratio*. The fuel ratio is the percentage of fixed carbon divided by the percentage of volatile matter as determined by the proximate analysis. In some borderline cases, classification is difficult, and cannot be based entirely on the fuel ratio, but the physical properties have to be considered as well. This difficulty is greater in classifying the lower ranks. Among the Western coals it is difficult to distinguish between lignite and subbituminous coal by chemical analysis alone. There is a fairly sharp demarkation between brown, woody, or amorphous lignite, and crystalline, shiny black, subbituminous coal. But this difference cannot be readily observed in the results of chemical analyses. Likewise, analyses do not always show distinctions between bituminous and subbituminous coals.

In Table 24 are given the fuel ratios for the various ranks of coal, together with the approximate range in B.t.u. per pound of combustible as determined from the proximate analysis. (Combustible is the wet coal minus the moisture and ash.)

The fuel ratios in Table 24 are from proximate analyses made

¹ *Professional Paper 100-A*, 1922, U. S. Geological Survey.
Bulletin No. 25, 1925, Virginia Geological Survey.

TABLE 24

FUEL RATIOS AND HEATING VALUES OF THE COMBUSTIBLE FOR THE VARIOUS RANKS OF COAL

RANK	FUEL RATIO	B.T.U. PER LB. OF COMBUSTIBLE*	REMARKS *
Anthracite.	10 to 99	14 800–15 200	Found in Penn., Col., Rd. Is., Wash., Va., & Alaska
Semianthracite. . .	5 to 10	15 000–15 500	Chiefly in Penn., Va., and Ark.
Semibituminous. . .	2.5 to 5	15 000–16 200	Found in Penn., W. Va., and Va.
Bituminous.	Below 2.5	13 500–15 800	Widely distributed East and West
Subbituminous. . .	Below 1.50	12 000–14 000	Western States
Lignite.	Below 1.50	10 000–13 000	Western and Northern States

* These items were supplied by the author. All others in the table are as used by the U. S. Geological Survey.

with an electric furnace. Analytical results for a given coal in different laboratories and at different times in the same laboratory if gas is the fuel used, do not always agree. The best modern practice is to use an electric furnace. All determinations of volatile matter in Government laboratories since 1915 have been made in an electric furnace of special design.¹ Analyses made in the electric furnace give a lower percentage of volatile matter than analyses in which gas flames were used, and results obtained in electric furnaces by different analysts agree satisfactorily.

Anthracite coal has a fuel ratio ranging from 99 to 10. It is a very hard coal having a high black luster. It is difficult to kindle, and burns with a short bluish flame. It burns without producing smoke or soot. It is an ideal domestic fuel, for which purpose it is used almost exclusively. The screenings, too small in size for domestic use, are mixed with other coals for central power station use in certain eastern cities having stringent smoke ordinances. Anthracite has a high carbon content and is very low in oxygen. It is considered to be the highest rank in the coal scale. Nearly all of it is mined in Pennsylvania, although limited measures are found in a few other states including Alaska, Colorado, New Jersey, and New Mexico.

¹ See *Bulletin 123*, U. S. Bureau of Mines, 1918, p. 7.

Semianthracite coal is also a hard coal but not quite so hard as anthracite. It will soil the hands slightly. It has a fuel ratio varying from 10 to 5. It kindles more readily than anthracite because of the higher content of volatile matter. It burns with a short yellow flame, is practically smokeless and produces no soot. When it burns it softens and swells slightly, but does not cake. It is used mostly for domestic purposes. It is used for power production only in the sizes which are too small for the domestic market, and it is used in cities where smoke ordinances are enforced. There is comparatively little semianthracite in this country.

Semibituminous coal is superior in rank to bituminous. It is unfortunate that a name for it indicating this fact was not chosen. But the name seems to be too well established by long use in the coal trade to be displaced. It has a fuel ratio varying from 5 to 2.5. It kindles easily, and burns with a long yellow flame. The fixed carbon content is high enough and the volatile matter low enough to give a nearly smokeless and sootless fire. The coal is generally minutely jointed and breaks up easily. In mining and transporting the coal, especially if it is rehandled, most of the lumps are broken into small sizes when it reaches the consumer. But the small sizes are just as valuable fuel as the lumps, provided they can be burned in a properly designed furnace equipped with suitable grates. It has the highest heating value of all the coals because its volatile hydrocarbon content approximates methane in composition. Semibituminous coal occupies less bunker space per unit of heat than any other coal. Largely for this reason, it is considered to be the best coal for marine use. When it burns it softens, swells considerably, and cakes. *Pocahontas coal* is one of the best known brands of this rank. It is mined in the Pocahontas fields of Virginia and West Virginia. It is much used in power production, and in making coke for metallurgical purposes.

The term *bituminous* is applied to the coal rank which constitutes by far the most important group of coals in the production of power. It applies to coals whose fuel ratio is below 2.5. This rank includes a large variety of coals from nearly all sections of

the United States and Alaska. Many attempts have been made to find some satisfactory basis upon which this rank could be subdivided, but no proposed scheme has met with general acceptance as yet. This rank contains both coking and non-coking coals. But just why one coal cokes and another will not is unknown. Both coking and noncoking coals have chemical analyses extending over about the same limits of variation; hence there is little possibility of establishing a rank for coking coals. Some have divided bituminous into an Eastern and a Western group. The United States Geological Survey has distinguished three divisions: *high-rank*, *medium-rank*, and *low-rank bituminous coals*. But just where the border lines are to be placed is difficult to determine, and is not agreed upon.

This rank also includes *cannel coals*. These coals contain more hydrogen and consequently more volatile matter than the other bituminous coals. They are very high in heating value, and burn with a long bright flame without fusing or coking. They are ideal to use in open grates, and have long been used for that purpose. They are also used to a limited extent to enrich manufactured illuminating gas. "Cannel coal owes its richness to the fact that it is composed almost entirely of the spores, spore cases, seed coats, and resinous or waxy products of such plants as lived at the time of the existence of the coal swamps."¹ The absence of woody material in such coal gives it a regular texture and grain that are not found in other coals.

The term *subbituminous* has been adopted by the United States Geological Survey for what has generally been called black lignite. Its maximum fuel ratio seems to be about 1.5. Most of the coal in this rank cannot be readily distinguished by general appearance from the lower grades of bituminous coals. Its chief distinguishing characteristic is that it slacks upon exposure to the air for any length of time. Bituminous coals practically never slack. The moisture content of subbituminous coal in the mine is very high. Repeated wetting and drying of the coal is liable to start spontaneous combustion, hence care has to be exercised in shipping and storing this kind of coal.

¹ See *Professional Paper 100-A*, U. S. Geological Survey, 1922, p. 6.

Those coals which are distinctly brown in color or contain woody, fibrous material, and are high in moisture (30 to 40 per cent) and oxygen are called *lignite*. They are intermediate in development between peat and subbituminous coal. The maximum value of the fuel ratio is apparently about 1.5. It parts readily with much of its moisture and slacks upon exposure to the weather. Spontaneous combustion is more likely to be started, thus greater care must be observed in storing it than even for subbituminous coal. Lignites are marketed almost entirely near the mine and are generally used for domestic fuel. They are mined at the present time only in those localities remote from the higher rank coals. There are enormous beds of lignite in the Dakotas and Texas. It will probably not become of great commercial importance until the higher rank coals are nearing exhaustion. The lignite beds are probably extensive enough to serve the United States for several hundred years as a source of heat and power after the higher ranks of coal are exhausted. This fuel appears to be especially well adapted to the manufacture of gas in gas producers.¹

116. Pulverized Coal. Pulverized coal has been used many years for fuel in rotary cement and lime kilns, in puddling furnaces, and in soaking pits of the steel industry. It is only during the past ten years that attention has been directed to its use in steam power plants. Within that short space of time it has become standard practice to burn the lower grades of coal in the pulverized form in the largest and most modern central power stations.

An extensive literature which has greatly aided the rapid development of the art of burning pulverized fuel has grown up. Only a few references to the more important papers can be given here.² The chief advantage of pulverizing coal is that it permits

¹ See *Economic Methods of Utilized Western Lignites*, Bulletin 89, U. S. Bureau of Mines, 1916.

² Haslam and Russell, *Fuels and Their Combustion*, 1926, pp. 427-466. *Symposium on Powdered Fuel*, Transactions, A.S.M.E., vol. 36, 1914, pp. 85-169. (Largely of historical interest now.)

Also Transactions, A.S.M.E., vol. 38, 1916, p. 1043; vol. 42, 1920, p. 255; vol. 47, 1925, p. 87.

Babcock & Wilcox Co., *Steam*, 36th ed., p. 233.

Proceedings National Electric Light Association, 1924, pp. 1499-1530; and 1927, pp. 1240-1358. This article contains a bibliography of 160 titles.

Transactions, A.S.M.E., vol. 50 (1928), FSP-papers, Nos. 27, 71, and 72.

the use of low-grade coal which is high in ash, moisture, and low in heating value. It also makes it possible to operate boilers at high ratings and with efficiencies somewhat higher than is possible with coal in coarse form and with grates or stokers. Its use is being rapidly introduced in the largest central power stations of the Middle West where the coal deposits are not of the highest quality. The firing control of pulverized coal is entirely automatic, as is the case for gas or oil. Since there is no heavy bed of fuel in the furnace, firing control may be so adjusted as to readily care for rapidly fluctuating-load demands.

One of the most important items to be looked after in preparing and handling pulverized coal is the moisture content. In the early days of its use in power plants, it was difficult to pulverize and transport coal containing more than two per cent of moisture. But as improved machinery has been developed and better methods of handling devised, coals containing as much as 9 per cent of moisture are pulverized and burned without previous drying.

Pulverized coal is usually prepared at the plant by one of two systems. In the central-storage system, preparation of the coal is made in a centrally located plant. The prepared coal is then stored in bins holding several days' supply. It is transported to the boilers as needed by means of screw conveyor or in a low-pressure air duct. In the unit system, each boiler or battery of boilers is equipped with an individual pulverizing plant located near the boiler, which prepares the fuel only as fast as it is consumed.

As the raw coal enters the plant, it is first passed through a magnetic separator, which removes tramp iron. If it is not already screenings, it is crushed to pass a 1-inch or 2-inch-mesh screen. If the storage system is in use, the moisture content is reduced in dryers to about 2 per cent. Unless this is done trouble arises due to the tendency of the moist, pulverized coal to pack and form lumps. Moreover, the spontaneous combustion hazard is greater for moist coal. Reducing the moisture content reduces the power required for pulverizing as well as increases the capacity of the pulverizers. The power consumption for pulver-

izing varies from 12 to 25 kilowatt-hours per ton of coal; depending upon the type of pulverizing equipment, and the condition of the coal. In modern practice using ball mills or impact type pulverizers with modern burners, coal of the fineness as stated in Table 25 is successfully burned.

TABLE 25
SIZE OF PULVERIZED COAL GRAINS

MESH OF SCREEN	PERCENTAGE OF COAL PASSED	MESH OF SCREEN	PERCENTAGE OF COAL PASSED
40	99	100	80 to 90
60	96 to 98	200	50 to 80
80	90 to 95		

The combustion of pulverized coal develops very high temperatures which cause rapid deterioration of furnace walls. The most successful preventative is the installation of water walls which consist of rows of water tubes extending vertically along the sides and back of the furnace. The walls form part of the water-circulating system of the boiler. Efficient combustion of pulverized coal seems to demand very large furnace volumes; thus making large boiler space necessary. In congested areas where land values are high, this feature adds considerably to the investment cost. However, much intensive research is being carried on to develop furnaces of smaller volume by producing more violent turbulence of furnace gases.

Advantages of pulverized coal are:

1. Poor and relatively cheap ranks and grades of coal may be readily burned with resulting high efficiency.
2. Boiler efficiencies are slightly higher with pulverized coal than with stokers or grates.
3. Automatic and instant control of the fires is possible, thus maintaining very uniform pressure and superheat for rapidly fluctuating-load demands.

Disadvantages encountered are:

1. Additional investment is necessary for the pulverizing equipment. Also the additional cost of the increased cubic content of the boiler room is an appreciable item.

2. It is not generally economical, as yet, to install powdered coal equipment in small plants.
3. Unless dust precipitators are installed, there is a continuous shower of ash in the vicinity of the plant, which may become objectional. Without precipitators, from 20 to 40 per cent of the ash is dissipated into the atmosphere.

The advisability of adopting pulverized coal for a new plant project can be settled only by a careful study of all the factors bearing upon the particular problem, each new plant being a special case requiring complete investigation by qualified engineers. It is doubtful if the savings and other advantages would in general justify scrapping equipment already in use in an established plant for the purpose of installing a new pulverized-coal system.

117. Heat-Treated Coal. For many years it has been realized that it is very wasteful of the fuel resources to burn raw coal in any kind of a furnace. In 1881 Sir William Siemens, the inventor of the open-hearth steel furnace, said, "I am bold enough to go so far as to say that raw coal should not be used for any purpose whatsoever, and that the first step toward the judicious and economical production of heat is the gas retort or the gas producer in which the coal is converted into gas, or gas and coke as in the case of our ordinary gas works."

This plan involves the partial consumption of the coal in coke ovens in which most of the volatile matter is driven off, and used in the form of gas for industrial and domestic purposes. The coke is then to be used for power and domestic as well as metallurgical purposes.¹ When heat treatment is applied to all bituminous and lower ranks of coal, the yield of by-products in the forms of ammonium sulphate and light oils is generally more than sufficient to repay the additional expense involved in carrying out the treatment. Besides, there is an increased efficiency of utilization of our fuel resources which is the most desirable and important result of the treatment. Heat treatment also provides a satisfactory solution for the troublesome smoke problem in large cities and industrial centers. As a health measure alone, the elimination of smoke would probably be worth the entire cost of heat treatment for all fuels below the rank of semianthracite coal.

¹ See U. S. Bureau of Mines, *Technical Paper 242*, 1919.

At the present time, about 14 per cent of all bituminous and lower ranks of coal mined is used in the form of coke for metallurgical purposes. About 80 per cent of the coke is produced in retort coke ovens. This percentage is constantly growing. It is but a comparatively short step to so prepare all coal. But this very desirable result will probably be obtained only by a slow process of evolution extending over several decades.

There are two processes of heat treatment in use, the *high-temperature* and the *low-temperature* processes.¹ The older method using beehive or retort coke ovens is the high temperature process in which temperatures from 1800° F. to 2200° F. are usual. The old beehive ovens are rapidly being discarded. A description of them and the method of their operation will not be attempted here.²

The by-product coke ovens are arranged in a series of rectangular cells placed side by side like ordinary building brick set on the long narrow side as a base. A system of gas passages is arranged between and around the cells for the purpose of supplying the heat necessary to carry on the coking process. The inside dimensions of the cells vary over a wide range, depending upon the type of oven, the nature of the coal to be coked, and the use to be made of the coke. The more usual dimensions are 30 to 40 feet long, 8 to 12 feet high, and about 16 inches wide at one end and about 20 inches at the other. The cells or ovens are charged ordinarily to within 12 or 15 inches of the top; thus holding from 8 to 15 tons of coal. Anywhere from 10 to 100 by-product ovens may be grouped together in a battery. A complete battery of ovens with all the auxiliary apparatus is an elaborate and costly brick structure.

The operation of either beehive or by-product ovens is an intermittent process. It requires about 20 to 28 hours to complete the coking process on a charge of coal in the by-product ovens and from 36 to 72 hours in the beehive. The ovens are "pulled" and recharged alternately so that the heat from the adjacent ovens help to start the new charge. Each ton of coal will give off during the coking process from 6000 to 15,000 cubic feet of a good grade of coal gas having an average heating value of 500 to 600 B.t.u. per cubic foot. In modern well-designed ovens utilizing the waste heat of the flue gases to preheat the air for combustion, about 40 per cent of the gas

¹ See Haslam and Russell, *Fuels and Their Combustion*, p. 646. Horace C. Porter, *Coal Carbonization*.

² See John Fulton, *Treatise on Coke*, pp. 145-198.

evolved is burned to heat the ovens. The remainder of the gas becomes available for domestic or power purposes outside the coke plant.

The maximum temperature employed in the relatively low-temperature carbonization process varies from 1000° F. to 1400° F. There are a number of low-temperature systems in use which differ as to details of equipment and mode of operation. The best process or the most suitable equipment with which to carry on the treatment has probably not yet appeared.¹ The chief disadvantage of this process is that the coal is heated slowly. This reduces or destroys the coking tendency. The result is a granular fragile fuel unsuited to metallurgical purposes. However, it is well adapted to use with stokers or for pulverizing. Before it can be used for domestic purposes it must be made into briquettes.

TABLE 26
ANALYSES OF COAL AND COKE

PROXIMATE ANALYSIS, PER CENT BY WEIGHT			ULTIMATE ANALYSIS, PER CENT BY WEIGHT		
	BY-PRODUCT OVENS *			BY-PRODUCT OVENS *	
	Coal	Coke		Coal	Coke
Moisture	8.07	7.15	Carbon	67.51	77.81
Volatile matter	34.66	2.18	Hydrogen	5.32	1.17
Fixed carbon	48.38	77.93	Oxygen	15.75	6.55
Ash	8.89	12.74	Nitrogen	1.49	.88
			Sulphur	1.04	.85
			Ash	8.89	12.74
Total	100.00	100.00	Total	100.00	100.00
			B.t.u. per lb.	12,020	11,493

* See *Coking of Illinois Coal in Koppers-Type Oven*, Technologic Paper No. 137, 1919, U. S. Bureau of Standards.

118. Fuel Oil. Coals furnish about 82 per cent of the power which is transformed from heat energy in the world. But the importance of fuel oil as a source of power is growing very rapidly because of the ease and convenience in handling and storing it, and because of the higher efficiency obtainable.

¹ See the *K.S.G. Process of Low-Temperature Carbonization*. Transactions, A.S.M.E., Vol. 50 (1928), No. 15, p. 17.

Approximately 18 per cent of the total power from heat today is derived from oil and gas.¹

Fuel oil is the residue which remains after the gasoline, kerosene, and the various lubricating-oil fractions have been removed from the crude petroleum by distillation. The United States Bureau of Mines have defined fuel oil as including "all oils which are not salable for some other special purpose at a higher price than that which prevails for oils to be sold as fuel oils to be burned under boilers." As the art of refining crude petroleum progresses, means are being found to break up or "crack" the heavier constituents of crude oil and thus produce much larger percentages of the lighter and more valuable oils. At the present time, it is commercially possible by use of cracking processes to convert from 50 to 75 per cent of crude petroleum into gasoline and kerosene. It is very likely that within the next few years, unless new and large pools of crude oil are discovered, that the rapidly increasing demands for automotive and Diesel engine fuels will render it economically impossible to burn fuel oil under steam boilers.²

Petroleum pools are found in small areas widely distributed over the earth's surface. At the present time about 75 per cent of the world's petroleum production comes from the United States and Mexico. However, there are indications that large undeveloped oil pools exist in the United States as well as in Russia, Turkey, China, British America, Alaska, and South America.

The physical properties of crude petroleum vary rather widely considering the fact that the chemical composition of oils from widely separated fields is surprisingly uniform. Based largely upon the physical properties, petroleum is classified into three broad groups: *paraffin-base*, *mixed-base*, and *asphaltic-base* oils.

1. Paraffin-base petroleum is a light, greenish-colored oil found largely in the central and eastern United States. Its specific gravity (water = 1.00) varies from approximately 0.82 to 0.92. From 20 to 80 per cent of gasoline may be distilled from it without

¹ See *Power Resources of the World*, 1929, by The World Power Conference.

² See *The Economic Status of Oil for Marine Fuel and Railway Practices in the Use and Conservation of Fuel Oil*. Transactions, A.S.M.E., vol. 50 (1928), FSP papers, Nos. 67 and 68.

cracking. It is so named because the residue after distilling off the light oils contains considerable paraffin wax.

2. *Mixed-base petroleum* is somewhat heavier and darker in color. It contains both paraffin and asphalt in varying proportions. This type of oil is found largely in the mid-continent and southern states, and in Mexico. Its specific gravity varies from 0.88 to 0.97, and the gasoline content in the uncracked oil varies from 10 to 25 per cent.

3. *Asphaltic-base oils* are heavy black oils containing from 5 to 15 per cent of gasoline. They are found in southwestern United States, Mexico, and Russia. Their specific gravity varies from 0.92 to 1.00. They contain considerable asphaltic material. Fuel oil for steam-boiler plants is derived largely from this group.

Table 27 contains representative analyses of the three classes of petroleum and some of the more important distillates.

TABLE 27
ANALYSES OF LIQUID FUELS

No.	KIND AND LOCALITY	SPECIFIC GRAVITY	ULTIMATE ANALYSIS PER CENT BY WEIGHT				
			C	H	O+N	S	Higher Heating Value B.t.u. per Pound
1	Crude oil, Eastern *	0.866	86.5	12.4	1.0	0.1	19 890
2	Crude oil, Mid-Continent *	0.933	86.0	11.2	2.1	0.7	19 440
3	Crude oil, Mid-Continent *	0.909	88.0	11.4	0.2	0.4	19 650
4	Crude oil, California ‡	0.950	86.7	11.5	1.2	0.6	18 730
5	Crude oil, Western *	0.970	86.7	11.4	0.8	1.1	18 480
6	Crude oil, Mexico ‡	0.912	82.9	12.2	2.1	2.8	18 495
7	Fuel oil, Mid-Continent *	0.955	83.3	10.2	3.7	2.8	18 427
8	Fuel oil, California ‡	0.940	85.7	11.3	2.4	0.6	18 500
9	Fuel oil, Western *	0.946	86.2	11.9	1.2	0.7	18 790
10	Kerosene †	0.793	85.3	14.6	0.1	..	19 890
11	Gasoline †	0.716	85.0	14.9	0.1	..	20 480
12	Alcohol, denatured †	0.818	47.2	12.6	40.2 §	..	11 630

* These analyses are taken by permission from G. F. Gebhardt's *Steam Power-Plant Engineering*, 6th ed., p. 60.

† See *Bulletin No. 43*, 1912, U. S. Bureau of Mines.

‡ These analyses are from private sources.

§ Alcohol contains no nitrogen.

An examination of Table 27 shows the rather interesting fact that the composition of petroleum and its products varies through a very narrow range considering the many hydrocarbon compounds found in oils from all the various fields widely scattered in all countries of the world. The extreme carbon variation is from about 82 to 88 per cent, and the hydrogen varies from about 10 to 15 per cent. In general the lighter oils contain the higher percentages of hydrogen, and are correspondingly low in carbon. Sulphur is, in a strict sense, a combustible, but it is an undesirable component of any fuel.

119. Colorific Power of Petroleum and Its Products. When two or more elements unite to form a chemical compound, as in the case of carbon and hydrogen in petroleum compounds, heat flow in one direction or the other takes place. During the synthesis of certain hydrocarbon compounds, such as methane (CH_4), and ethane (C_2H_6), heat is liberated. When these compounds are broken down again into their elements, as in the process of combustion, the heat of formation must be restored, thus reducing the effective heat of combustion. In the case of certain other compounds, among them being ethylene (C_2H_4), benzene (C_6H_6), anthracene ($\text{C}_{14}\text{H}_{10}$), and others, the reverse is true.¹ Petroleums from the different oil fields of the world contain a wide variety of hydrocarbons. Those from the eastern part of the United States are rich in saturated hydrocarbons which are called the paraffin ($\text{C}_n\text{H}_{2n+2}$) series. The asphaltic-base oils of California, Texas, and Mexico contain high percentages of unsaturated compounds in most part belonging to the ethylene (C_nH_{2n}) series. The heavy oils of Russia contain naphthenes (cyclohexane and its homologues). Cyclohexane has the formula C_6H_{12} .² For these reasons it is impossible to devise a universally satisfactory formula which may be used to compute the calorific power of petroleums from their ultimate analysis. However, many attempts have been made to develop such a formula.³ Below are given two formulas which are probably as

¹ A. T. Lincoln, *Physical Chemistry*, pp. 418-422.

² C. K. Tinkler and F. Challenger, *Chemistry of Petroleum*.

³ See W. Inchley in *The Engineer*, Vol. 3, p. 155 (London); and Sherman and Krapff, *Journal of the American Chemical Society*, Oct., 1908.

satisfactory as any. The results obtained from their use will usually be found to be within ± 3 per cent of the value found by calorimeter. In rare instances, however, the error may be as great as ± 10 per cent. The formulas given in equations (269) and (270) are intended to be used in computing the heating values of heavy fuel oils whose specific gravities usually lie between 0.980 and 0.930, corresponding respectively to about 12 and 20.5 degrees by the Baumé hydrometer.

$$(269) \quad U = 14\,600\,C + 53\,000\,H,$$

$$(270) \quad U = 17\,200 + 90\,B.$$

In these equations

U = higher calorific value of one pound of dry oil in B.t.u.

C = percentage of carbon in the oil by ultimate analysis (used as a decimal),

H = percentage of hydrogen in the oil by ultimate analysis used (as a decimal),

B = the gravity of the oil in degrees Baumé at 60° F.

The ultimate analysis of the oil must be available before equation (269) can be used. It is necessary to measure only the gravity of the oil with a Baumé hydrometer, a comparatively simple operation, to determine the heating value by equation (270), since the results obtained by either equation are only approximate.

It will probably be found just as satisfactory to use only equation (270).

The heating value of fuel oils of Western United States and Mexico usually varies from 18 000 to 19 000 B.t.u. per pound of dry oil, with an average value not far from 18 500 B.t.u. Unless definite calorimeter results are available, it is probably as accurate to use the average value stated above as to attempt to compute the value. The heating values of the lighter Eastern paraffin-base oils varies from 19 000 to 20 000 B.t.u., the heating value of kerosene from 19 500 to 20 000 B.t.u., and that of commercial gasolene from 20 000 to 21 000 B.t.u. Crude oils from Russia and Java are reported as having values as high as 21 000 B.t.u. per pound.

120. Gaseous Fuel. Combustible gases are the most highly esteemed fuel available, owing to the high efficiency obtainable in their utilization, the ease of instantaneous control, saving of labor, and cleanliness. There are two classes, *natural gas* and *manufactured gas*.

Appreciable amounts of power are generated from natural gas in only a few favored localities, generally in or near oil fields. Such localities, however, are especially favored in so far as a cheap and convenient fuel is a factor in generation of industrial power. In the aggregate, the total quantity of power produced from natural gas is small, probably not over 2 per cent of the total. Large quantities of natural gas, however, are used for domestic purposes and in industrial processes.

Manufactured gas is also very largely used for domestic and industrial purposes, and relatively a small quantity is used to produce power. In some industries, such as the manufacture of steel, however, there is available, as a by-product of blast furnace and coke oven operation, considerable quantities of gas. At the present time, much of this gas is used under boilers and in internal-combustion engines about steel plants.

If all the raw coal used of the semibituminous and lower ranks were treated by a carbonization process, large quantities of gaseous fuel would become available for power generation. The carbonization process could best be carried on near the coal mines with the development of superpower plants in the immediate vicinity to economically convert the heat energy of the gas into power. The coke or carbonized-coal residue which would be a smokeless fuel could then be shipped to far points for power and domestic purposes. This is an ideal development which should materialize in the not far distant future.

Table 28 contains analyses of representative samples of natural gas and each type of manufactured gas. The column marked C_nH_{2n} series gives the percentages of what is generally termed *illuminants* because they are the elements which give gas its illuminating property when burned in an open fish-tail burner. The heating value, unless especially noted, is the higher heating value for one cubic foot of gas when measured at a pressure of 30

inches of mercury and at a temperature of 60° F. This is the usual manner of specifying gas values and quantities. The footnote numbers below the table correspond to the gases as numbered in the table.

TABLE 28
ANALYSES OF REPRESENTATIVE GASES *
(Per Cents by Volume)

No.	KIND OF GAS	LO-CALITY	C ₂ H ₂ SE-RIES	C ₂ H ₆	C ₂ H ₄	CH ₄	H ₂	CO	CO ₂	O ₂	N ₂	B.T.U. PER CU. FT.
1	Natural	Pa.			18.5	80.8					0.7	1140
2	Natural	Ind.			6.2	86.8		0.8			6.2	984
3	Natural	Texas			10.4	51.3			0.1		38.2	700
4	Natural	Calif.			5.6	94.2			02.0			1049
5	Natural	Calif.			22.6	77.4						1178
6	Natural	Calif.			20.2	62.7			15.5	0.2	1.4	884 L
7	Coal gas	Pa.	3.0			31.8	51.6	6.5	1.1	0.8	5.2	611
8	Water gas	Calif.				2.2	50.0	39.7	4.6	0.4	3.1	322
9	Oil gas	Calif.	3.0			30.0	38.3	13.8	8.0	0.2	6.7	530
10	Producer		1.4			3.0	10.4	26.8	4.0	0.2	54.2	163
11	Coke oven		4.2	1.2		35.5	48.0	5.1	1.3	0.5	4.2	679
12	Blast furnace	Chi-cago				0.2	3.9	25.5	13.4	O ₂ + N ₂ = 57.0		96

* NOTE: L indicates lower heating value; all the others are higher values.

References:

- 1, 2, and 3. See *Technical Paper No. 109*, 1915, U. S. Bureau of Mines.
- 4 and 5. Pacific Gas and Electric Co.
6. See *Transactions, A.S.M.E.*, vol. 34 (1912), p. 233.
7. See *Bulletin No. 6*, 1911, U. S. Bureau of Mines.
8. See *Pacific Service Magazine*. Pacific Gas and Electric Co., 1913.
9. Test by author, May, 1917.
11. See *Transactions, A.S.M.E.*, vol. 30 (1908), p. 408.
12. See *Transactions, A.S.M.E.*, vol. 32 (1910), p. 440; and vol. 46, p. 213.

121. Combustion and the Products of Combustion. From the point of view of engineers, combustion is the chemical change which takes place when a fuel combines rapidly with oxygen. It is always accompanied by an evolution of heat energy which may be transformed into useful mechanical work. The oxygen necessary to carry on combustion is taken from the atmosphere. Hence if a quantitative study of combustion is to be made, the chemical composition of the air must be known. Air is not a chemical compound, but it is a mechanical mixture of elemental gases approximately as shown in Table 29.

Besides the constituents listed in Table 29, air contains small but varying amounts of water vapor and traces of helium, krypton, neon, xenon, ammonia, nitrous acid, nitric acid, sulphurous and sulphuric acids.

TABLE 29
CONSTITUENTS OF AIR
(Per Cents by Volume)

Oxygen.....	20.65
Nitrogen.....	78.40
Argon.....	0.90
Carbon dioxide.....	0.03
Total.....	99.98

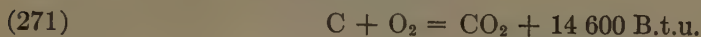
For the purposes of the engineer in studying combustion, air is considered to be composed of only oxygen and nitrogen. This assumption can be made safely without changing materially the results and conclusions arrived at from a study of flue-gas analyses. The analysis of air, upon that basis, is assumed to be as shown in Table 30.

TABLE 30
CONSTITUENTS OF AIR

ELEMENT	VOLUME PER CENT	WEIGHT PER CENT
Oxygen.....	20.9	23.2
Nitrogen.....	79.1	76.8
	100.0	100.0

The theoretical quantity of air required to completely oxidize a pound of any fuel may be readily computed from the fundamental chemical equations, the ultimate analysis of the fuel, and Avogadro's law. It may be recalled that Avogadro's law is, *At the same pressure and temperature, equal volumes of different gases contain the same number of molecules.* Hence the weights of equal volumes of gases are proportional to the number of molecules. The converse of this statement is also true: namely, the weights of unit volumes of different gases at the same pressure and temperature are proportional to their atomic weights or to their molecular weights. Therefore, molecular formulas must be used in all equations, for some gases are compounds and

others are elements. Combustion equations for carbon, hydrogen, and sulphur are given below.



Molecular weights $12 + 32 = 44$.

For 1 lb. of carbon $1 + 2\frac{2}{3} = 3\frac{2}{3}$.

Hence to burn one pound of carbon completely and without excess oxygen left over, $2\frac{2}{3}$ pounds of oxygen are required. The resulting products of combustion would be $3\frac{2}{3}$ pounds of carbon dioxide gas accompanied by the liberation of 14 600 B.t.u., which are made available for useful purposes.

One pound of air, as shown in Table 30, carries 0.232 lbs. of oxygen. Hence to just burn completely one pound of carbon without excess air, there would be required a supply of air equal to

$$(272) \quad 2\frac{2}{3} \div 0.232 = 11.49 \text{ lbs.}$$

The products of combustion would now include the nitrogen which must necessarily accompany the required weight of oxygen, namely,

$$11.49 \times 0.768 = 8.82 \text{ lbs.}$$

Hence the theoretical products of combustion when one pound of carbon is completely burned without excess air are

Carbon dioxide	3.667 lbs.
Nitrogen	8.823 "
Total	12.49 "

The total weight of the products is one pound greater than the weight of the air supplied.

Proceeding in the same manner for carbon burned to CO, for hydrogen, and for sulphur, the following equations and values result. When one pound of carbon is burned to CO,



$$(273) \quad 24 + 32 = 56,$$

$$1 + 1\frac{1}{3} = 2\frac{1}{3}.$$

Hence one pound of carbon burned to CO requires

$$(274) \quad 1\frac{1}{3} \div 0.232 = 5.75 \text{ lbs. of air.}$$

When one pound of carbon in the form of CO is burned to CO₂



$$(275) \quad \begin{array}{l} 24 + 32 = 56, \\ 1 + 1\frac{1}{3} = 2\frac{1}{3} \end{array}$$

The air required will be

$$(276) \quad 1\frac{1}{3} \div 0.232 = 5.75 \text{ lbs.}$$

When one pound of hydrogen is burned

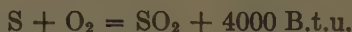


$$(277) \quad \begin{array}{l} 4 + 32 = 36, \\ 1 + 8 = 9. \end{array}$$

It is thus seen that one pound of hydrogen when completely burned without excess air requires 8 pounds of oxygen, 9 pounds of water vapor are formed, and 62 000 B.t.u. are made available for useful work. One pound of hydrogen requires, for complete combustion without excess oxygen,

$$(278) \quad 8 \div 0.232 = 34.48 \text{ lbs. of air.}$$

When one pound of sulphur is burned with just sufficient oxygen for theoretically complete combustion without excess, the equations are



$$(279) \quad \begin{array}{l} 32 + 32 = 64, \\ 1 + 1 = 2. \end{array}$$

Hence one pound of sulphur requires one pound of oxygen and produces 2 pounds of sulphur dioxide accompanied by the release of 4000 B.t.u. The weight of air required is

$$(280) \quad 1 \div 0.232 = 4.31 \text{ lbs.}$$

122. The Combustion Problem. There are two general types of combustion problem which the engineer is frequently called upon to solve. First, it may be required to predict from computations and the ultimate analysis of the fuel the products of

combustion by weight, by volume, and in terms of per cents by weight and by volume for different quantities of air supplied. The solution of this type of problem is necessary in determining the size and height of a suitable stack or stacks for a new boiler plant. Second, and by far the more frequent problem, is to determine the weight and volume of the stack gases leaving a furnace per pound of fuel, and to determine whether the air supply is adequate or greatly in excess of that required and how much. The data required for the solution of the latter problem are the ultimate analysis of the fuel and the analysis of the flue gases leaving the furnace. The ultimate analysis of the fuel is generally reported in percentages by weight. The flue-gas analysis is generally reported as obtained, that is, in percentages by volume. Hence one of the most important steps in the solution is the conversion of the gas-analysis results from the volume to the weight basis and vice versa. For this purpose, the use of the following equations based upon Avogadro's law is necessary:

Let

M_1, M_2, M_3 , etc. = the respective weights of the various component gases, lbs.,

V_1, V_2, V_3 , etc. = the respective volumes of the various constituents in cu. ft.

n_1, n_2, n_3 , etc. = respectively the number of molecules per unit of volume of each constituent gas,

m_1, m_2, m_3 , etc. = the respective molecular weights of each constituent of the gas,

p_1, p_2, p_3 , etc. = the percentages by weight of the respective constituents,

q_1, q_2, q_3 , etc. = the percentages by volume of the respective constituents,

$\sum M$ = Summation of all the respective weights, M_1, M_2, M_3 , etc.; and in like manner for all the other quantities used, the sign $\sum$ placed before the symbol indicates the summation of all like quantities relating to the various constituents of the gas.

Then

$$(281) \quad p_1 = \frac{100 M_1}{\sum M}; \quad p_2 = \frac{100 M_2}{\sum M}; \quad p_3 = \frac{100 M_3}{\sum M}, \quad \text{etc.},$$

$$(282) \quad q_1 = \frac{100 V_1}{\sum V}; \quad q_2 = \frac{100 V_2}{\sum V}; \quad q_3 = \frac{100 V_3}{\sum V}, \quad \text{etc.},$$

But

$$(283) \quad \begin{cases} M_1 = n_1 m_1; & M_2 = n_2 m_2; & M_3 = n_3 m_3, & \text{etc.}, \\ M_1 \sim m_1 v_1 \sim m_1 q_1. \end{cases}$$

And in the same way for the respective volumes

$$(284) \quad V \sim n; \quad V \sim \frac{M}{m}.$$

Then

$$(285) \quad p_1 = \frac{100 M_1}{\sum M} = \frac{100 n_1 m_1}{\sum n m} = \frac{100 m_1 V_1}{\sum m V} = \frac{100 m_1 q_1}{\sum m q},$$

and

$$(286) \quad q_1 = \frac{100 V_1}{\sum V} = \frac{100 n_1}{\sum n} = \frac{100 \frac{M_1}{m_1}}{\sum \frac{M}{m}} = \frac{100 \frac{p_1}{m_1}}{\sum \frac{p}{m}}.$$

In the same manner expressions for p_2 , p_3 , q_2 , q_3 , and so on may be written.

Equation (285) gives weight percentages in terms of weights, in terms of volume, and in terms of volume percentages, respectively. The expressions in (286) give volume percentages in terms of volume, in terms of the number of molecules, and in terms of weight percentages.

The use of these equations in the solution of the first problem outlined at the beginning of this section can best be explained by carrying through the solution of an illustrative problem.

ILLUSTRATIVE PROBLEM

From the ultimate analysis of a bituminous coal shown in Table 31, compute the theoretical weight of air required to burn completely and without excess air one pound of the coal; also determine the products of combustion in percentages by weight and by volume when the air supply is 1.5 times the theoretical requirement.

TABLE 31
ULTIMATE ANALYSIS OF A BITUMINOUS COAL
Dry Coal Equals 100 per Cent by Weight *

Carbon.....	79.3 per cent
Hydrogen.....	4.3
Nitrogen.....	1.4
Oxygen.....	3.4
Sulphur.....	2.1
Ash.....	9.5
Total.....	100.0

* B.t.u. per pound of dry coal = 14,000.

For purposes of boiler-room computations, engineers assume that all of the oxygen in the coal as shown by the analysis is combined with hydrogen in the proportion to form water of crystallization. This assumption is probably not strictly true, but it is sufficiently accurate to justify its use. On that basis, only a portion of the hydrogen can be considered as available for combustion. The only combustible constituents of the coal are carbon, the uncombined hydrogen, and the sulphur. Then from (272), (278), and (280) the theoretical weight of air required is computed as follows:

$$(287) \quad A = 11.49 C + 34.48 \left(H - \frac{O}{8} \right) + 4.31 S.$$

In (287), A represents the pounds of air theoretically required to just burn completely and without excess one pound of fuel. The symbols C , H , O , and S represent respectively the pounds of carbon, hydrogen, oxygen, and sulphur in one pound of fuel as indicated by the ultimate analysis. Substituting these values from Table 31 in equation (287), one obtains

$$(288) \quad A = 11.49 \times .793 + 34.48 \left(0.043 - \frac{0.034}{8} \right) + 4.31 \times 0.021,$$

$$(289) \quad A = 10.54 \text{ lbs. of air required.}$$

Let A_f represent the pounds of air actually supplied per pound of fuel. Then

$$(290) \quad A_f = 10.54 \times 1.5 = 15.81.$$

The percentage weights of the various products of combustion when one pound of coal is burned in 15.81 pounds of air are computed by use of (281) as indicated in Table 32.

TABLE 32
PRODUCTS OF COMBUSTION

GAS	WEIGHT OF THE PRODUCT IN POUNDS	$\frac{M_1}{\Sigma M}$, ETC.	PERCENT-AGE WEIGHTS
CO ₂	$3\frac{2}{3} \times 0.793 = 2.91 = M_1$	$p_1 = \frac{2.91 \times 100}{16.72} =$	17.5
H ₂ O	$9 \times 0.043 = 0.39 = M_2$	$p_2 = \frac{0.39 \times 100}{16.72} =$	2.3
SO ₂	$2 \times 0.021 = 0.04 = M_3$	$p_3 = \frac{0.04 \times 100}{16.72} =$	0.2
N ₂	$15.81 \times 0.768 + 0.014 = 12.16 = M_4$	$p_4 = \frac{12.16 \times 100}{16.72} =$	72.7
O ₂	$(15.81 - 10.54) 0.232 = \frac{1.22}{16.72} = \Sigma M = M_5$	$p_5 = \frac{1.22 \times 100}{16.72} =$	7.3
			100.0

The products, of combustion including water vapor in percentage by volume, are obtained from the weight percentages given in Table 32 by the use of the fourth righthand member of equation (286) as shown in Table 33.

TABLE 33
PRODUCTS OF COMBUSTION IN PERCENTAGES BY VOLUME

GAS	A NUMBER PROPORTIONAL TO VOLUME	$\frac{100 \frac{p_1}{m_1}}{\Sigma \frac{p}{m}} =$	PERCENT-AGE BY VOLUME
CO ₂	$\frac{p_1}{m_1} = \frac{17.5}{44} = 0.3978$	$q_1 = \frac{39.78}{3.3533} =$	11.9
H ₂ O	$\frac{p_2}{m_2} = \frac{2.3}{18} = 0.1278$	$q_2 = \frac{12.78}{3.3533} =$	3.8
SO ₂	$\frac{p_3}{m_3} = \frac{0.2}{64} = 0.0031$	$q_3 = \frac{0.31}{3.3533} =$	0.1
N ₂	$\frac{p_4}{m_4} = \frac{72.7}{28} = 2.5965$	$q_4 = \frac{2.5965}{3.3533} =$	77.4
O ₂	$\frac{p_5}{m_5} = \frac{7.3}{32} = 0.2281$	$q_5 = \frac{0.2281}{3.3533} =$	6.8
	$\Sigma \frac{p}{m} = \text{Total} = 3.3533$	Total.....	100.0

In Tables 32 and 33, the water vapor of combustion is included. The usual method of volumetric analysis of stack gases determines only the constituents of the dry gases. The water vapor is condensed and is not measured. The results in percentages by weight and by volume for *dry gases* may be obtained from the results of Tables 32 and 33 as follows. The sum of the percentages by weight in Table 32 when water vapor is omitted is 97.7. By dividing each percentage by 0.977 (omitting water vapor) the sum is again made equal to 100.0 per cent. Similarly for the results in Table 33 the sum of the percentages by volume are made and equal 100.0 per cent with water vapor excluded. These steps are carried out in Table 34.

TABLE 34
PRODUCTS OF COMBUSTION IN DRY GAS

GAS	PERCENTAGES BY WEIGHT	PERCENTAGES BY VOLUME
CO ₂	$\frac{17.5}{0.977} = 17.9$	$\frac{11.9}{0.962} = 12.4$
SO ₂	$\frac{0.2}{0.977} = 0.2$	$\frac{0.1}{0.962} = 0.1$
N ₂	$\frac{72.7}{0.977} = 74.4$	$\frac{77.4}{0.962} = 80.4$
O ₂	$\frac{7.3}{0.977} = 7.5$	$\frac{6.8}{0.962} = 7.1$
	Total. 100.0	100.0

When flue gases are analyzed, the sulphur-dioxide content is absorbed and measured with the carbon dioxide. It is usually so small a percentage of the total gas, as illustrated in Tables 32, 33, and 34, that this error is negligible. For this reason it is customary, unless extreme scientific accuracy in the results is desired, to ignore the sulphur content of the fuel in computing the products of combustion in the same manner that the presence of water vapor is ignored.

123. Flue-Gas Analysis. There are two types of flue-gas analyzers in general use by engineers: the stationary automatic-recording type, and the portable hand-operated indicating Orsat type (so named after the inventor). Structurally, the recording instrument is complicated and delicate. Its first cost is high, and it requires somewhat skilled attention to maintain it in good operating adjustment.

In most of its forms, only one of the constituent gases, CO_2 or O_2 , are determined.¹

The Orsat instrument,² one form of which is illustrated in Figs. 44 and 45, consists of a graduated burette *A* connected by means of a branched manifold *J* to three or four absorption pipettes *B*, *C*, *D*, and *E*, and a leveling-bottle *F* filled with brine or mercury which serves

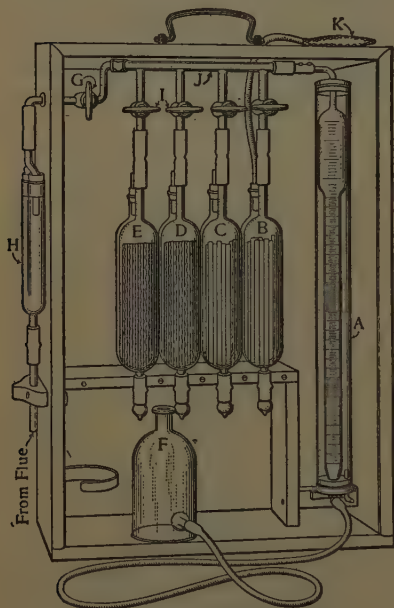


FIG. 44.—ORSAT ANALYZER

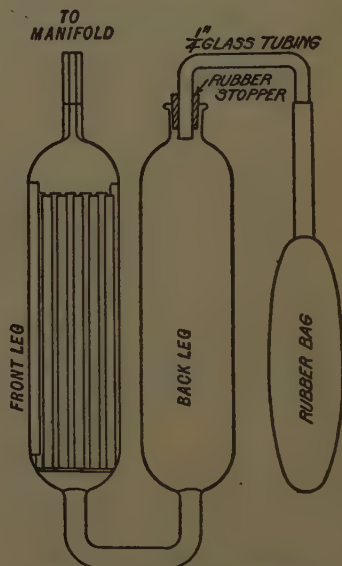


FIG. 45.—ORSAT PIPETTE

as a pump to move the gas sample. There are two sizes of Orsat in use. The standard instrument is provided with a 100-cc. burette. There is a half-size instrument provided with a 50-cc. burette. The half-size instrument is for use where rapidity is more important than a high degree of precision in the results. The burette is water-jacketed to reduce to a minimum errors due to changing temperature during an analysis which requires from 5 to 15 minutes to complete. The absorption pipettes are twin glass vessels, each of about 125 cc. capacity connected at the bottom by a U-shaped tube as shown in

¹ See G. F. Gebhardt, *Steam Power-Plant Engineering*, p. 848, for descriptions of the Republic CO_2 recorder, Uehling CO_2 records, and the Engelhard gas analyzer. Also *Bulletin 91* 1916, United States Bureau of Mines.

² See *Bulletin 97*, 1915, U. S. Bureau of Mines, for complete description, methods of use, and flue-gas analysis calculations for the Orsat.

Fig. 45. The front leg of each pipette contains a nest of small glass tubes which give greater surface of contact between the gas and the reagent, thus hastening the analysis. A rubber bag or a water seal is attached to the back leg of all pipettes except the first (or *B* pipette) to protect the chemical against rapid deterioration by contact with the atmosphere.

Pipette *B*, next to the burette, contains about 150 cc. of a 30 per cent solution, by weight, of potassium hydroxide in water. It absorbs the carbon dioxide and has no effect upon the other components of the flue gas. This solution has great affinity and capacity for CO_2 . One volume of the solution is capable of absorbing as much as 40 volumes of carbon dioxide. The second pipette *C* contains a solution of pyrogallic acid (photographic pyro) in a 30 per cent solution of potassium hydroxide (about 5 grams of pyro in each 100 cc. of KOH) for absorption of oxygen. One cubic centimeter of this solution has the capacity to absorb about 10 cubic centimeters of oxygen. The third and fourth pipettes contain an ammoniacal solution of cuprous chloride. Or a solution of cuprous chloride in hydrochloric acid may be used instead. This reagent absorbs the carbon monoxide of the gas. It also has the ability to absorb oxygen and carbon dioxide. Therefore these two gases must be completely removed before the sample is introduced into the cuprous chloride pipettes. Two cuprous chloride pipettes are provided, and are used one after the other to make the absorption complete. Copper wire is inserted in the nests of small glass tubes in the cuprous chloride pipettes in order to maintain the solution in the reduced or cuprous form.

The burette and pipettes are connected by a branched manifold having a very small bore and consequently negligible capacity. Glass stopcocks *I* are provided between each pipette and the manifold. However, some engineers prefer pinchcocks and short connecting pieces of small rubber tubing for valves because the glass cocks give much trouble by sticking and breaking if not properly lubricated. A three-way cock *G*, provided with a colored spot to indicate the direction of the passages open, closes the outer end of the manifold. The sample of gas is taken into and discharged from the Orsat through the three-way cock.

The leveling-bottle *F*, filled with mercury or NaCl brine, is connected by about 3 feet of pure gum rubber tubing to the lower end of the burette. Mercury is preferable because all gases are insoluble in

it, but salt water is more convenient to handle and gives more accurate leveling adjustments for reading gas volumes in the burette. The mercury or brine serves as a piston to displace the gas in the burette as the leveling-bottle is raised or lowered. The liquid also serves to control the pressure of the gas at the instant of reading its volume in the burette. It will be observed that the analysis of gases by the Orsat is a *volumetric analysis*.

Carbon dioxide, oxygen, and carbon monoxide are determined by absorption in their respective pipettes in the order named. The principal constituents of flue gas are the three gases mentioned above and nitrogen. Small amounts of water vapor and sulphur dioxide are always present. The most of the water vapor is condensed and cannot be measured in the Orsat. It may be determined by computation from the ultimate analysis of the fuel. The quantity of sulphur dioxide is always so small as to be insignificant as regards combustion economy. It is absorbed and measured with the carbon dioxide, and no attempt is made to separate them. The nitrogen is always found by subtracting from 100 per cent the sum of the percentages of carbon dioxide, oxygen, and carbon monoxide. Since the atmosphere contains 21 per cent of oxygen by volume and it can only be replaced by the carbon dioxide, carbon monoxide, and water vapor, the sum of the three constituents measured (CO_2 , O_2 , and CO) can never exceed 21 per cent. It is assumed that all the remaining gas is nitrogen, which assumption may be considerably in error. This assumption automatically transfers any errors which may have been made in carrying out the analysis to the nitrogen item.

An analysis of flue gas made by use of the Orsat instrument consists in taking the sample from the flue; transferring it to the apparatus; absorbing the various constituents in the respective reagents in the pipettes; and measuring the volume of the sample before and after each absorption. Care must be exercised to obtain a truly representative sample, and skill and vigilance are required to prevent errors in performing the analysis. The principal sources of error are deterioration of the reagents and leakage into or out of the instrument through the stop cocks and various rubber connections.

The second type of problem mentioned in § 122, namely, to determine whether too much or just sufficient air is being supplied to the furnace and to determine the amount of heat lost up the stack, may now be taken up.

When the fuel whose analysis is given in Table 31 (page 201) was

burned in a boiler furnace, the analysis of the resulting flue gases made by the Orsat instrument was found to be as given in Table 35.

TABLE 35
FLUE GAS ANALYSIS
(Percentages by Volume in the Dry Gases)

Carbon dioxide.....	11.6
Oxygen.....	7.3
Carbon monoxide.....	0.4
Nitrogen.....	80.7
Total.....	100.0

$$\text{The nitrogen} = 100.0 - (\text{CO}_2 + \text{O}_2 + \text{CO}).$$

Based upon the assumption that the fuel is completely consumed, the weight of air supplied and the weight of dry stack gases per pound of fuel may be computed by use of equations (291) to (299) inclusive. Let

A = the theoretical pounds of air required to burn completely one pound of dry fuel,

A_c = the pounds of air actually supplied per pound of carbon burned,

A_f = the pounds of air actually supplied per pound of dry fuel burned,

C = the weight of carbon in one pound of dry fuel as shown by the ultimate analysis,

D = the dilution coefficient,

E = the excess air supplied over and above that theoretically required,

M_c = the pounds of dry gases produced per pound of carbon burned,

M_f = the pounds of dry gases produced per pound of fuel burned,

CO_2 = carbon dioxide, percentage by volume,

O_2 = oxygen, percentage by volume,

CO = carbon monoxide, percentage by volume,

N_2 = nitrogen, percentage by volume.

By equation (283), the total weight of the dry gases is proportional to

$$(291) \quad 44 \text{ CO}_2 + 32 \text{ O}_2 + 38 \text{ CO} + 28 \text{ N}_2.$$

On the same basis, the weights of carbon in the carbon dioxide and in the carbon monoxide are proportional to

$$(292) \quad 44 \text{ CO}_2 \times \frac{12}{44} = 12 \text{ CO}_2,$$

and

$$(293) \quad 28 \text{ CO} \times \frac{12}{28} = 12 \text{ CO}.$$

Hence the weight of dry gases per pound of carbon burned is equal to $(291) \div [(292) + (293)]$. Then

$$(294) \quad M_c = \frac{44 \text{ CO}_2 + 32 \text{ O}_2 + 28 (\text{CO} + \text{N}_2)}{12 \text{ CO}_2 + 12 \text{ CO}},$$

or

$$(295) \quad M_c = \frac{11 \text{ CO}_2 + 8 \text{ O}_2 + 7 (\text{CO} + \text{N}_2)}{3 (\text{CO}_2 + \text{CO})},$$

and

$$(296) \quad M_f = M_c \times C.$$

The weights of air supplied per pound of carbon and per pound of dry fuel may be computed, approximately, from the nitrogen content of the dry gases. By weight, the atmosphere contains 76.8 per cent nitrogen. Then the total weight of a given volume of atmospheric air is proportional to

$$(297) \quad \frac{28 \text{ N}_2}{0.768},$$

and

$$(298) \quad A_c = \frac{\frac{28 \text{ N}_2}{0.768}}{12 (\text{CO}_2 + \text{CO})} = \frac{3.038}{\text{CO}_2 + \text{CO}}.$$

$$(299) \quad A_f = A_c \times C.$$

The weight of air obtained by (298) is likely to be in error by a small amount because all fuels contain some nitrogen, either in a free state or in some combined form which may break down in the furnace, and the percentage of nitrogen is obtained by difference; hence all the errors of the analysis are automatically transferred to the nitrogen item.

The values stated in Table 31 (page 201) and Table 35 (page 207) may be used in the above equations to work out an illustrative problem. From equations (295) and (296) the weight of dry gases is

found to be

$$(300) \quad \begin{aligned} M_c &= \frac{11 \times 11.6 + 8 \times 7.3 + 7 (0.4 + 80.7)}{3 (11.6 + 0.4)} \\ &= \frac{753.7}{36.0} = 20.94 \text{ lbs.} \end{aligned}$$

$$(301) \quad M_f = 20.94 \times 0.793 = 16.61 \text{ lbs.}$$

Likewise by equations (298) and (299) the weight of air is

$$(302) \quad A_c = \frac{3.038 \times 80.7}{12} = 20.42 \text{ lbs.}$$

$$(303) \quad A_f = 20.42 \times 0.793 = 16.19 \text{ lbs.}$$

The weight of dry gases per pound of fuel should come out approximately one pound greater than the weight of air per pound of fuel because the air passing through the furnace has had added to it by the combustion process only the pound of fuel. Because of the various unavoidable errors of the analysis, such as measuring the SO_2 in with the CO_2 , neglecting the presence of water of combustion, neglecting the nitrogen in the fuel, and the determination of the nitrogen in the gases by difference, the difference between the weights of dry gases and air never comes out exactly right. However, if the discrepancy is large, one may be certain that a serious error has been made in performing the analysis or in making the computations.

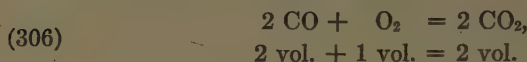
It is often convenient to know the quantity of air supplied to a furnace in terms of the theoretical requirement. On that basis, the minimum requirement for perfect combustion may be considered as unity. The number expressing the actual supply in terms of this unit is known as the dilution coefficient, symbolized in equation (307) by the letter D . The nitrogen in the stack gases as determined by analysis may be taken as a measure of the actual air supply, and the oxygen as shown by the analysis to be a measure of the excess air supplied. The relationship between the volumes of nitrogen and oxygen in the air is expressed by

$$(304) \quad \frac{\text{O}_2}{\text{N}_2} = \frac{20.9}{79.1},$$

or

$$(305) \quad \text{N}_2 = \frac{79.1}{20.9} \text{O}_2 = 3.785 \text{ O}_2.$$

Then 3.785 O₂ represents the nitrogen that is included with the excess air passing into the furnace provided there is no carbon monoxide formed. On a volume basis



Hence, for each volume of CO produced, one-half volume more of oxygen would be required to burn it to carbon dioxide. Therefore

$$(307) \quad D = \frac{N_2}{N_2 - 3.785 (\text{O}_2 - \frac{1}{2} \text{ CO})},$$

and

$$(308) \quad E = (D - 1).$$

in which N₂, O₂, and CO represent the respective percentages of nitrogen, oxygen, and carbon monoxide by volume as determined by analysis. The dilution coefficient *D*, as determined by (307), represents the ratio of the actual air supply to that theoretically required for complete combustion. This method of computing it is recommended in the *Power Test Code* of the American Society of Mechanical Engineers.

Substituting the necessary values from the flue gas analysis of Table 35 (page 207), one obtains

$$(309) \quad D = \frac{80.7}{80.7 - 3.785 (7.3 - 0.2)} = 1.499,$$

and

$$(310) \quad E = 1.499 - 1.000 = 0.499.$$

But since the percentage of nitrogen is found by difference and may be considerably in error, the following method of computing the dilution coefficient based upon the oxygen content of the flue gas is more accurate provided the ultimate analysis of the fuel is available. The excess weight of air is proportional to

$$\frac{32 (\text{O}_2 - \frac{1}{2} \text{ CO})}{0.232}$$

And the excess weight of air per pound of carbon in the fuel is then

$$(311) \quad E_c = \frac{\frac{32 (\text{O}_2 - \frac{1}{2} \text{ CO})}{0.232}}{12 (\text{CO}_2 + \text{CO})} = \frac{11.5 (\text{O}_2 - \frac{1}{2} \text{ CO})}{\text{CO}_2 + \text{CO}}.$$

The excess weight of air per pound of fuel is

$$(312) \quad E_f = \frac{11.5 \text{ C}(\text{O}_2 - \frac{1}{2} \text{ CO})}{\text{CO}_2 + \text{CO}},$$

and

$$(313) \quad D = \frac{A + E_f}{A}.$$

Substituting the appropriate values from the fuel and gas analyses given on pages 201 and 207 in the above equations one obtains

$$(314) \quad E_f = \frac{11.5 \times .793 (7.3 - 0.2)}{12.00} = 5.40 \text{ lbs.}$$

On page 201 the value of A is found to be 10.54. Hence

$$(315) \quad D = \frac{10.54 + 5.40}{10.54} = \frac{15.94}{10.54} = 1.51.$$

Compare this value of D with the value in (309). The discrepancy may be considered as due to the errors in determining the percentage of nitrogen. The value 1.51 is probably the more nearly correct. For the same reason the total weight of air supplied per pound of fuel, as found by use of equations (311), (312), and (313), is probably more nearly correct than the value from equations (297), (298), and (299). From (313) it is seen that

$$(316) \quad A_f = A + E_f,$$

and

$$(317) \quad A_f = 10.54 + 5.40 = 15.94 \text{ lbs.}$$

Compare this result with the value in (303), page 209.

It is not possible under the very strenuous conditions of intense combustion in modern boiler furnaces to burn fuel completely without some excess air. There is not time enough for each molecule of oxygen to find an appropriate molecule of fuel and combine with it because of the imperfect mixing of fuel particles and oxygen. From 20 to 50 per cent of excess air is required for coal burned on grates or automatic stokers if the amount of carbon monoxide formed is to be maintained below the economic limit. Fuels in the form of pulverized coal, oils, or gas may be burned with as little as 5 per cent excess air without producing appreciable quantities of carbon monoxide. An unduly large excess air supply means that the heat loss up the stack becomes very large. The heat carried out into the atmosphere in the dry chimney gases per

pound of fuel burned may now be computed, provided the room and stack temperatures are known.

$$(318) \quad L_g = \bar{c}_p(t_g - t_a)M_f$$

where

L_g = B.t.u. carried away in the dry stack gases per pound of dry fuel burned (a loss of heat),

M_f = the pounds of dry stack gases per pound of dry fuel,

t_g = the temperature of the stack gases, degrees F.,

t_a = the temperature of the air entering the furnace, degrees F.,

$\bar{c}_p$ = the mean specific heat of the stack gases at constant pressure.

The value, 0.24, for $\bar{c}_p$ has been arbitrarily chosen for all stack gases and for all temperature ranges met in ordinary boiler operation by the American Society of Mechanical Engineers as stated in the *Power Test Code*. This value has become thoroughly established by long usage. If a scientifically accurate value is desired, it must be recomputed for each new temperature range and for every analysis of stack gases.

For the purposes of a sample problem, take the weight of dry gases as found in (301), page 209, and assume the respective temperatures as shown below; then one obtains

$$(319) \quad L_g = 0.24 (500 - 70) 16.61 = 1714 \text{ B.t.u.}$$

The heating value of one pound of the dry coal is given in Table 31 (page 201) as 14 000 B.t.u. Hence the loss in the dry gases is

$$\frac{1714 \times 100}{14\,000} = 12.24 \text{ per cent.}$$

PROBLEMS

1. The coal analyses, page 213, are taken from *Bulletin No. 123* of the United States Bureau of Mines. All numbers are percentages by weight.

Compute the higher and lower heating values per pound of wet coal by Dulong's formula for each of these coals.

2. Compute the higher and lower heating values by means of De Paepe's modification of Goutal's formula.

3. Determine and state the rank in which each of the five coals whose analyses are given on page 213 belong.

4. Neglecting the sulphur content, compute the dry products of combustion in percentages by weight and by volume for Coal No. 4, page 213, when the value of the dilution coefficient is 1.5.

5. Repeat Problem 4 when the value of the dilution coefficient is 1.2.

ULTIMATE ANALYSES
(Dry Coal 100 per Cent)

ELEMENT	1	2	3	4	5
Carbon	71.0	82.5	60.5	77.0	73.0
Hydrogen	5.2	4.3	3.5	5.2	5.1
Sulphur	0.7	2.4	0.4	3.0	0.6
Nitrogen	1.6	1.3	0.8	1.5	1.6
Oxygen	10.3	1.6	19.2	5.5	13.2
Ash	11.2	7.9	15.6	7.8	6.5
Total	100.0	100.0	100.0	100.0	100.0

PROXIMATE ANALYSES
(Wet Coal 100 per Cent)

ITEM	1	2	3	4	5
Moisture	7.1	2.5	36.6	3.0	11.9
Volatile matter	37.1	16.6	23.2	36.7	36.8
Fixed carbon	45.4	73.2	30.3	52.8	45.6
Ash	10.4	7.7	9.9	7.5	5.7
Total	100.0	100.0	100.0	100.0	100.0
B.t.u. by calorimeter	11 716	14 114	6 136	13 516	11 234

6. Coal having the composition shown by analysis No. 1, above, gave the following flue-gas analysis in per cents by volume when burned in a boiler furnace: CO_2 , 9.3; O_2 , 10.2; CO , 0.0; N_2 , 80.5. (a) Compute the pounds of air supplied per pound of wet coal and the dilution coefficient. (b) Compute the pounds of dry stack gasses per pound of wet coal.

7. Determine the weight and volume of air required to burn completely without excess one cubic foot of natural gas whose analysis is given by No. 1, Table 28, page 195. Also determine the dry products of combustion in per cents by weight and by volume when there is no excess air supplied.

8. Repeat Problem 6 for fuel oil, analysis No. 9, Table 27, page 191, but having the following flue-gas analysis in per cents by volume: CO_2 , 12.2; O_2 , 3.8; CO , 0.0; N_2 , 84.0.

9. Compute the higher and lower heating values of fuel oil from analysis No. 7, Table 27, page 191, by use of equations (263), (264), (269), and (270).

10. Compute the B.t.u. and the per cent loss up the stack per pound of wet fuel for the conditions in Problems 6 and 8 when the room and stack temperatures are respectively 80°F . and 500°F .

11. Compute the various percentages by weight and by volume of the dry products of combustion for the fuel oil of Problem 8 if the dilution coefficient had been 1.75.

12. Determine the weight and volume of free air required to burn one cubic foot of water gas whose analysis is given by No. 6, Table 28, page 195, when the dilution coefficient is 1.10. Also determine the composition of the wet products of combustion in per cents by weight and by volume.

13. Compute the lower heating value of the water gas of Problem 12.

14. Why are molecular formulas used instead of atomic formulas in solving combustion problems?

15. Neglecting the presence of sulphur, compute the dry products of combustion in per cents by volume for the coal whose analysis is No. 1, page 213, when the following weights of air are supplied per pound of wet coal: 5.42, 9.50, 15, 25, and 50 pounds. In the cases of insufficient air supply for complete combustion, assume that all the hydrogen burns to H_2O first; then with the remaining air the carbon is next burned to CO before the formation of CO_2 begins. Plot these results in the form of curves.

16. Compute the percentages by weight for the various elements in the gases listed in Table 28, page 195.

CHAPTER XIII

STEAM BOILERS

124. General. Steam boilers are pressure vessels designed to transmit, with the smallest possible loss, the heat generated in the furnace into the water and steam contained within the boiler. The form and general design of the various types of steam boilers are substantially the same to-day as they were nearly fifty years ago, with the one exception of the bent water-tube type which came out in 1888. The introduction of high pressures and highly superheated steam has made changes necessary in the structural details, but the fundamental features remain as introduced half a century or more ago.¹ However, great improvements in the design of the furnace have been made in recent years. Automatic stokers have been developed and perfected. The use of pulverized coal is being generally introduced in large central-power stations since about 1920. The use of pulverized coal on a large scale is undoubtedly an improvement which has successfully passed the experimental stage in the art of burning the poorer grades of coal. The use of pulverized coal permits of instant and automatic control of the rate of combustion in accordance with rapidly changing load conditions. The introduction of automatic stokers and pulverized coal have been largely responsible for increasing the size of boiler units from the 600 or 800 horsepower with a maximum steam-generating capacity of about 50 000 lbs. per hr. of about 1910 to about 4000 boiler horsepower with a maximum capacity of one million pounds per hour of the present time.

(A builders' rated boiler horsepower is considered to be 10 square feet of active boiler heating surface. Active boiler heating

¹ See Terrell Croft, *Steam Boilers*, pp. 5-20, for a brief historical sketch on the evolution of the steam boiler. More extended historical treatments will be found in: Thomas Tredgold, *The Steam Engine*; N. P. Burch, *Practical Treatise on Boilers*, 1873; R. H. Thurston, *A Manual of Steam Boilers*, 1901; and F. J. Rowan, *The Practical Physics of the Modern Steam Boiler*, which is devoted almost exclusively to the water-tube boiler.

surface is defined as being all that metal surface of the boiler over which the hot gases of combustion flow with appreciable velocity on one side, and which is in contact with water on the other side. The gas side of the metal is the area considered.)

The realization of high economy in the generation of power from fuels depends primarily upon the correct arrangement of the boiler surfaces and upon the form and proportions of the boiler furnace.¹ The item of fuel is the major portion of the total operating expense of a power plant. In spite of the highly developed furnaces and automatic machinery to feed fuel into them, it still requires skill and good judgment, the result of long experience, combined with eternal vigilance on the part of the boiler-room force, to secure uniformly high boiler-plant efficiencies.

Experience indicates that modern boilers of various sizes and types are capable of about the same output of steam per pound of coal burned, provided the furnace is properly designed and skillfully operated. However, the types of modern boilers differ greatly in respect to the space occupied, weight, unit cost, and adaptability to particular conditions of location and operation.

There is a strong tendency toward standardization of boiler design and construction induced by state and national legislation, and by the codes of engineering societies. These laws and codes have become desirable or necessary as safety measures. The *Construction Boiler Code* of the American Society of Mechanical Engineers exercises an important influence upon design and construction of modern boilers. This Boiler Code has been incorporated into the laws concerning the design, inspection, and operation of boilers in many states. The specifications of the United States Navy and those of the United States supervising inspectors of the Department of Commerce control, or at least strongly influence, the design and construction of all boilers operated at sea under the American flag. Other countries have their boiler inspection boards whose standards are on a high plane, and in many respects similar to those of the United States.

¹ See E. T. Haslam and R. P. Russell, *Fuels and Their Combustion*, 1926, for a more complete treatment on fuels, combustion, furnaces, and stokers.

125. Boiler Classifications. Elaborate and extended classifications of steam boilers have been proposed from time to time.¹ But these classifications are often confusing because of overlapping of the various divisions. For sake of simplicity and clearness, only two broad groups will be differentiated as shown in Table 36, first, as to the location of the furnace, and second, as to the path of the products of combustion.

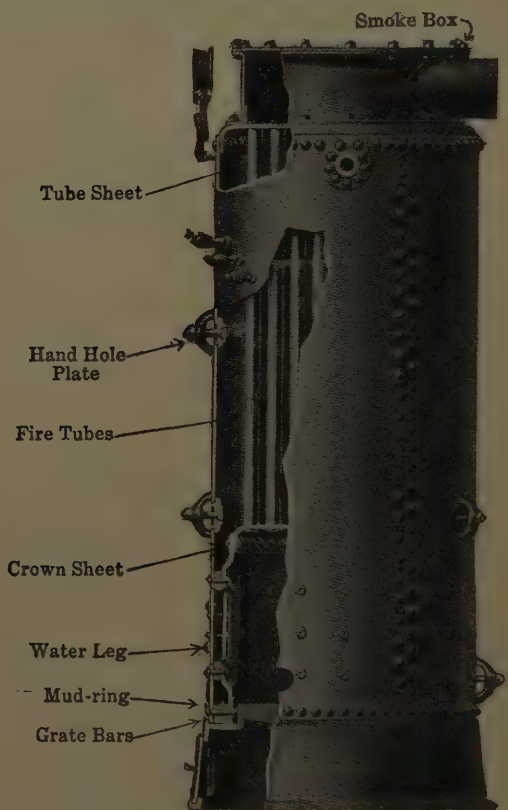
TABLE 36
CLASSIFICATION OF MODERN BOILERS

I	{	Internally fired	{	Vertical fire tube
			{	Scotch marine
			{	Locomotive
	{	Externally fired	{	Horizontal return tubular
			{	Water tube
II	{	Fire tube	{	Horizontal return tubular
			{	Scotch marine
			{	Vertical
			{	Locomotive
	{	Water tube	{	Longitudinal-drum, straight-tube
			{	Cross-drum
			{	Inclined-drum
			{	Vertical-drum
			{	Straight-tube
			{	Bent-tube

126. Vertical Boilers. In Fig. 46 is shown a sectionalized view of the vertical style of fire-tube boiler with internal fire box. It consists of an outside cylindrical shell which varies in size from 2 feet in diameter and 4 feet tall to approximately 8 feet in diameter and 15 feet tall. A cylindrical fire box 6 or 8 inches smaller in diameter is built into the lower end of the shell, being held in place by means of stay bolts, which are screwed and riveted to the two surfaces. The top of the fire box is closed by the crown sheet, which is drilled to receive the lower ends of the tubes. The top end of the shell is closed by another sheet drilled to receive the top ends of the tubes. The boiler is usually set on a cast-iron base containing the ash pit. The tubes, which are

¹ See R. H. Thurston, *A Manual of Steam Boilers*, p. 19; Edgar MacNaughton, *Elementary Steam-Power Engineering*, p. 16.

generally 2 inches in outside diameter and vary in number from 24 to about 300, depending upon the size of the boiler, are expanded into the crown and header sheets by rolling, and then the ends are beaded over. To some extent, the tubes serve as stays to brace the crown sheet and head.



(Titusville Iron Works Co.)

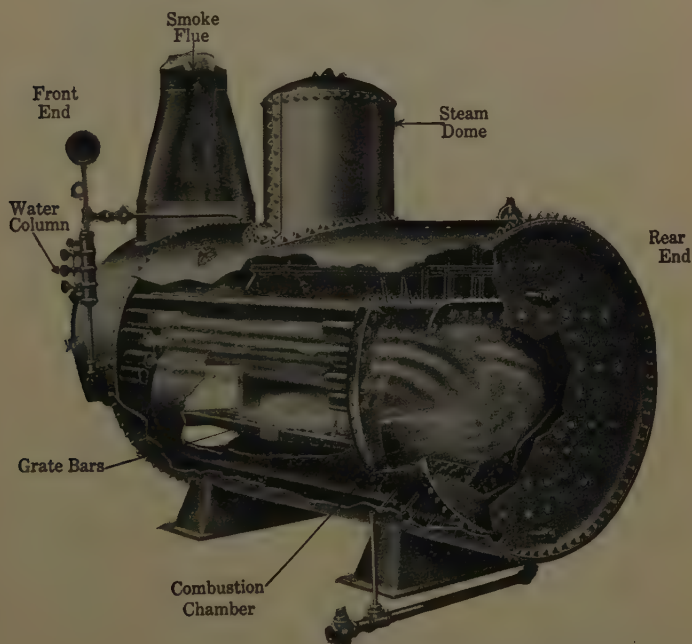
FIG. 46.—VERTICAL BOILER

The chief characteristics of the vertical boiler which enable it to survive are the simple self-contained structure (no masonry furnace setting is required), small floor space, portability, rapid generation of steam, low first cost. Besides they may be operated in many states and cities without a licensed engineer unless a specified pressure is exceeded.

Their chief disadvantages are low efficiency, poor water circulation, and slight overload capacity.

They are used mostly to furnish steam for hoisting engines, logging engines, laundries, bakeries, and other small plants in which steam is required in industrial processes. They are ordinarily built in capacities varying from 2 or 3 horsepower to 100 horsepower. Occasionally a 200 horsepower vertical boiler is built.

127. Scotch Marine Boilers. Views of Scotch marine boilers are shown in Figs. 47 and 48. They consist of a cylindrical, riveted shell varying from 6 to 20 feet in diameter and from 8 to



(Murray Iron Works Co.)

FIG. 47.—SCOTCH MARINE BOILER

12 feet long. The lower part of the shell contains from one to four corrugated-steel furnaces, riveted, at the front end, to flanged openings in the boiler-head; and at the rear end, they are riveted to the front wall and tube sheet of the combustion cham-

ber into which the furnaces open. The furnaces are cylindrical and vary from 3 to 4 feet in diameter and are about 2 feet shorter than the boiler shell. Because of their corrugations they are not usually braced by stays. The furnaces are surrounded by water, and they are fired internally. For coal, grate bars are placed

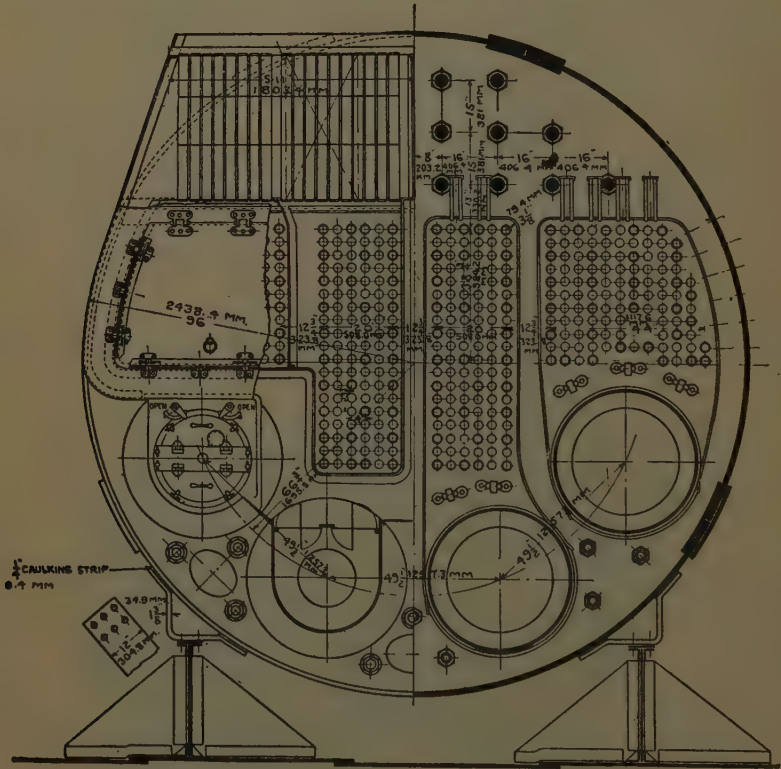


FIG. 48A.—SCOTCH MARINE BOILER—TRANSVERSE SECTION

across the front end of the furnaces slightly below the horizontal diameter. Above the furnaces, and leading from the combustion chamber forward to the front head, is a nest of fire tubes, usually $2\frac{1}{2}$ inches in external diameter (boiler tubes are generally designated by the external diameter). The hot gases of combustion pass through these tubes on their way to the smoke flue. Above the nest of tubes, the front and rear boiler heads are tied together

by through stays. The back wall and crown sheet of the combustion chamber are secured in place by stay bolts through the back head and the boiler shell. This type of boiler may be made double-ended, eliminating one head between two boilers, thus securing greater capacity per unit of weight and space occupied.

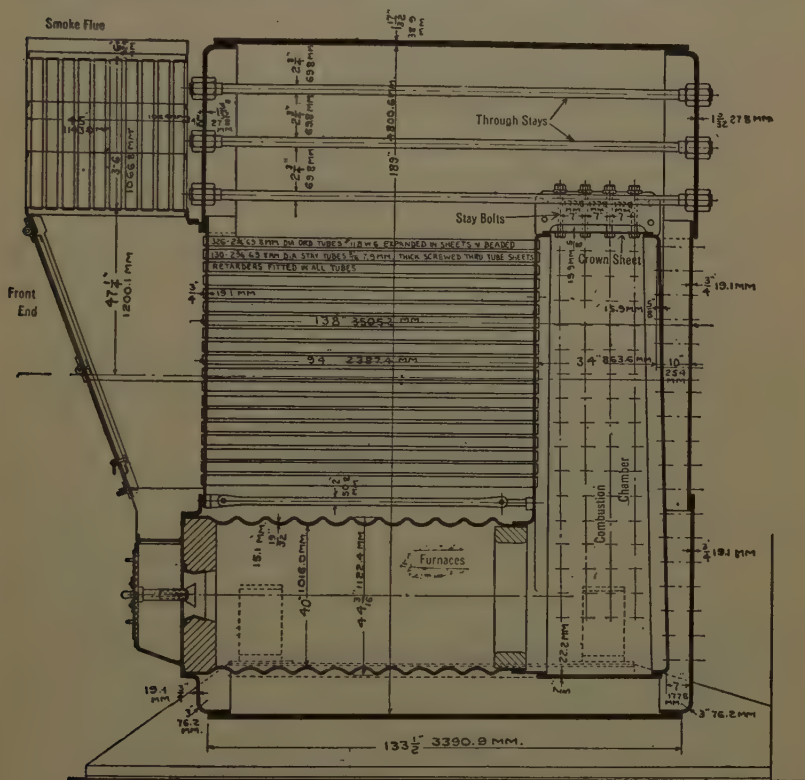


FIG. 48B.—SCOTCH MARINE BOILER—LONGITUDINAL SECTION

The Scotch type of boiler has been evolved, very largely, to meet the space and weight requirements of the mercantile marine. It is self-contained, requires no brick setting, and can be operated at high ratings without liability of damages. But the marine type of water-tube boiler, Fig. 57, is rapidly displacing the Scotch boiler because of its lighter weight, greater capacity per cubic foot of space occupied, and reduced accident liability.

128. Locomotive-Type Boilers. The general appearance and form of the locomotive type of boiler is shown in Fig. 49. It contains an internal fire box of rectangular shape inside the rear end of the boiler shell and separated from it 4 to 6 inches by braces and stay bolts. There are two general designs in use, the open-bottom and the closed-bottom fire box as shown in the figures. In the closed-bottom style, the fire box is entirely surrounded by water; thus eliminating the necessity for any masonry setting. About a third of the boiler's length forward the shell merges into a cylindrical barrel. The fire tubes are nested entirely within the barrel portion of the shell. The sheets forming the

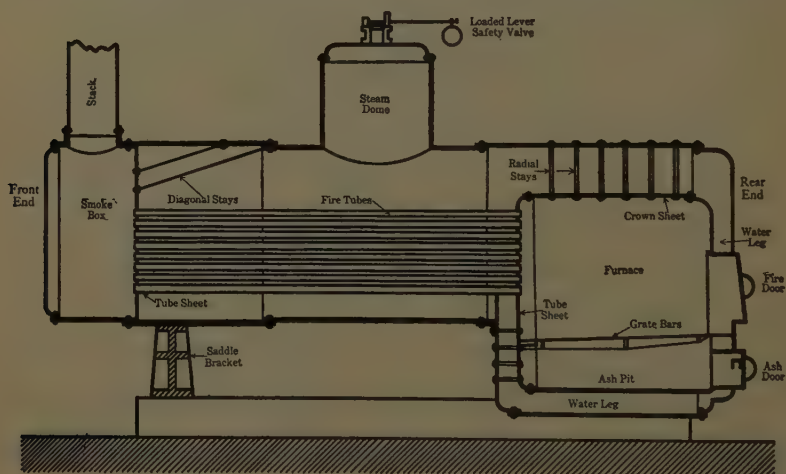


FIG. 49.—LOCOMOTIVE-TYPE BOILER

front end of the fire box and the front header are drilled to receive the tube ends, which are 2 to 3 inches in diameter for railway locomotive boilers, and 3 to 4 inches in diameter for stationary boilers. A smoke-box extension to which the stack is attached is riveted to the front end of the boiler shell. The hot products of combustion pass directly from the fire box through the tubes to the smoke box. The shell is filled with water to within 4 or 5 inches of the top, thus submerging the crown sheet of the fire box to a depth of a few inches. A steam dome is generally made a feature of the locomotive-type boiler, although it may be omitted

from the larger boilers and replaced by a dry pipe longitudinally disposed in the top of the shell. (See Fig. 64 for one form of dry pipe.)

This type of boiler is designed and built for steam pressures as high as 450 pounds per square inch, and boilers to operate at a pressure of 1700 pounds are being developed.¹ But for small stationary service the pressure seldom exceeds 200 pounds per square inch. For small-plant service where a semiportable, self-contained boiler is desired, the sizes range from 10 to 150 boiler horsepower. Railroad-locomotive boilers usually vary in size from 300 to 1000 boiler horsepower.

The outstanding advantages of the locomotive-type boiler are: portability, internal self-contained furnace, large capacity for their size and weight, and moderately high efficiency.

The disadvantages are: complicated structure, large flat surfaces requiring much stay bracing, difficulty of keeping clean inside, poor water circulation, high explosive risk, and rather high first cost.

129. Horizontal Return-Tubular Boilers. The horizontal return-tubular boiler is also known as the *fire-tube boiler* because the hot products of combustion pass through the tubes. This type of boiler, Figs. 50 to 52 inclusive, consists of a cylindrical shell closed at the ends by headers drilled to receive the ends of the nest of fire tubes which extend throughout the entire length of the shell. The boiler is placed in a horizontal position over external furnace walls constructed of ordinary brick with linings of insulating brick and fire brick.

The products of combustion impinge first upon the lower outside surfaces of the shell, then flow to the back end of the furnace. Here the hot gases are directed upward by the rear arch of the setting (Figs. 51 and 52) into the tubes through which they pass on their way to the smoke box and stack located at the front end. Thus the extent of surface over which the hot gases pass is considerably greater than is the case with internally-fired boilers. The boiler is usually supported over the furnace on lugs or

¹ Transactions A.S.M.E. R.R., vol. 51.

brackets riveted to the shell. In case the brackets rest directly on the brickwork, the rear brackets are generally mounted on rollers to permit of expansion and contraction due to temperature changes. The modern and better method is to support the boiler independently of the setting on a steel gallows frame as shown in Fig. 52.

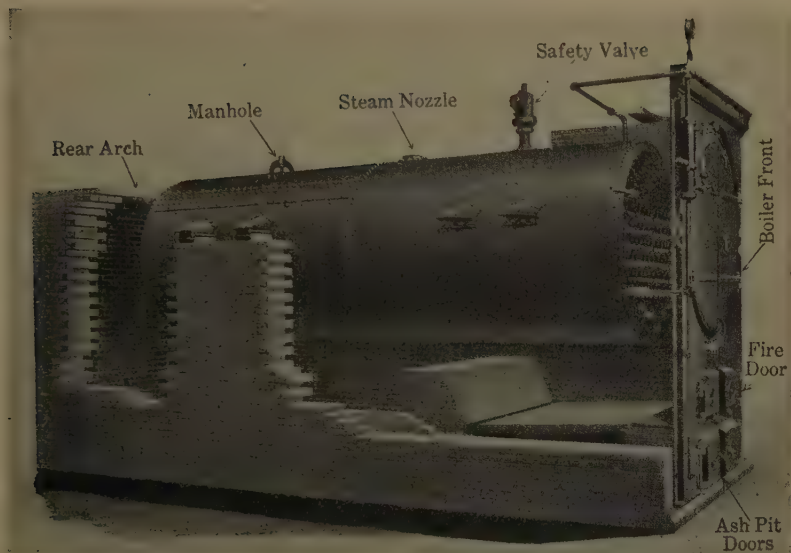
Horizontal return-tubular boilers are built in various sizes with shells varying from about 3 feet in diameter and 8 feet long containing 20 or 24 3-inch tubes and rated at about 15 horsepower to a shell diameter of 8 or 9 feet, 20 to 24 feet long and containing



FIG. 50.—HORIZONTAL RETURN-TUBULAR BOILER

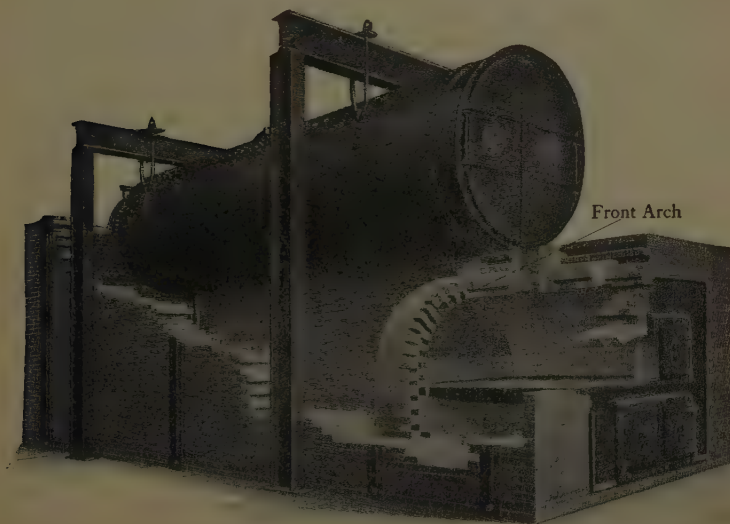
100 to 200 3-inch or 4-inch tubes. The corresponding maximum rating is about 500 boiler horsepower. But it is seldom that a horizontal return-tubular boiler is built larger than 7 feet in diameter and 350 horsepower rating. The majority of boilers in use throughout the United States belonging to this type vary in capacity from 40 or 50 horsepower to 200 horsepower. The steam pressure at which these boilers are operated rarely exceeds 150 pounds per square inch.

The chief advantages of the multitubular boiler are: large evaporative capacity in proportion to its space requirements, rapid circulation of the thin layers of water between the tubes, and low first cost per unit of heating surface.



(Erie City Iron Works)

FIG. 51.—HORIZONTAL RETURN-TUBULAR BOILER AND SETTING

FIG. 52.—HORIZONTAL RETURN-TUBULAR BOILER WITH DUTCH OVEN
AND GALLOWS FRAME

The more outstanding disadvantages are: the danger from explosion is rather great, as shown by the records of the past; the riveted circumferential seams of the shell must be exposed directly to the hot gases in the furnace, a condition which eventually results in weakening the seams; deposits of scale on the interior surfaces are difficult to remove; and a large masonry setting is necessary.

For the purpose of burning special fuels, such as sawmill refuse, bagasse, and spent tan bark, special settings with extra large furnaces and combustion chambers are required, as shown in Fig. 52. This setting is known as the Dutch-oven type. In some localities, where the cost of fuel is very high, steel-encased settings are highly profitable because the infiltration of cold air which reduces boiler and furnace efficiency can be entirely stopped.

Because of its low first cost and simplicity of operation, the vast majority of small and moderate-sized industrial plants of the country requiring up to 200 or 300 horsepower are using horizontal return-tubular boilers.

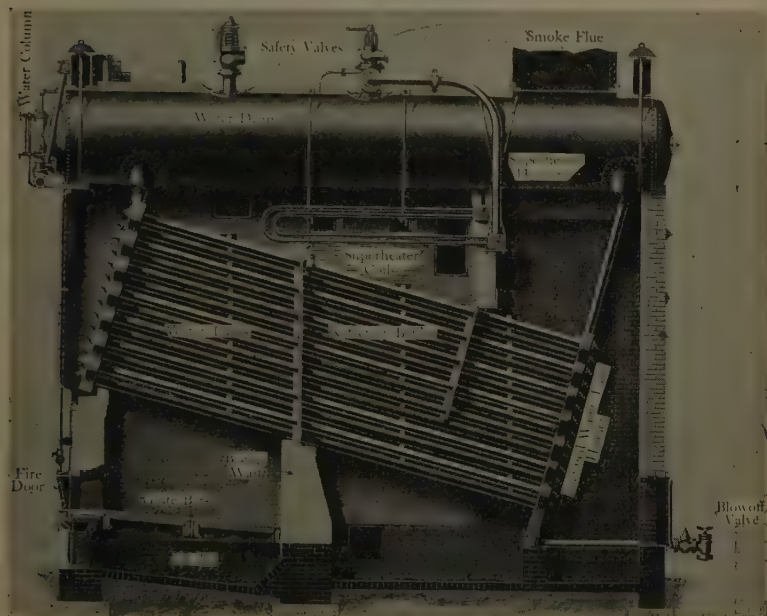
130. Water-Tube Boilers. The accumulated experience of more than 200 years in building and operating steam pressure-vessels indicates that a good boiler must fulfill certain specifications, the more important of which are enumerated below.

1. Accomplish maximum heat transfer from the hot products of combustion into the water and steam within the boiler.
2. Deliver a minimum of heat to the atmosphere.
3. Present a maximum of heating surface with minimum weight of metal.
4. Possess maximum inherent strength and safety due to form and arrangement.
5. Possess maximum steam-generating capacity with minimum space requirements.
6. The structure should be made up of simple forms conveniently accessible for cleaning and repairs.

To meet these specifications best, it has been found that the following secondary considerations should be met:

- (a) There should be maximum circulation of water inside the boiler.

- (b) There should be maximum turbulence and mixing of hot gases as they sweep over the heating surfaces.
- (c) There should be minimum thickness, and proper disposition of the heating surfaces consistent with safety.
- (d) There should be sufficient steam and water capacity to prevent undue fluctuation of water level and steam pressure with varying load demands.

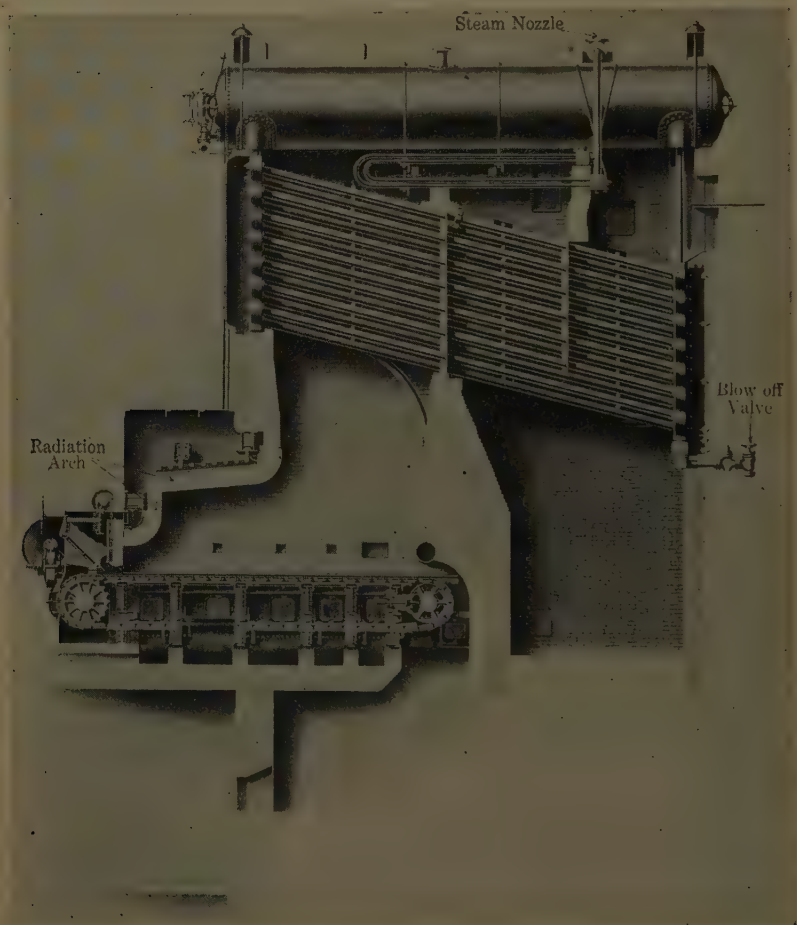


(The Babcock and Wilcox Co.)

FIG. 53.—BABCOCK AND WILCOX WATER-TUBE BOILER AND SETTING FOR HAND FIRING

Some of the above-mentioned requirements are conflicting, and the best possible compromise must be found and accepted. If the law of the survival of the fittest is valid, the design which most nearly fulfills all these requirements seems to be some form of water-tube boiler. The chief characteristic of this type of boiler is that the hot gases pass over the outside and the water is within the tubes, reversing the arrangement found in the horizontal return-tubular boiler.

131. Babcock and Wilcox Water-Tube Boilers.¹ Sectionalized views of this style of boiler are shown in Figs. 53 to 59 inclusive. The boiler as shown in Figs. 53 and 54 consists of a horizontal



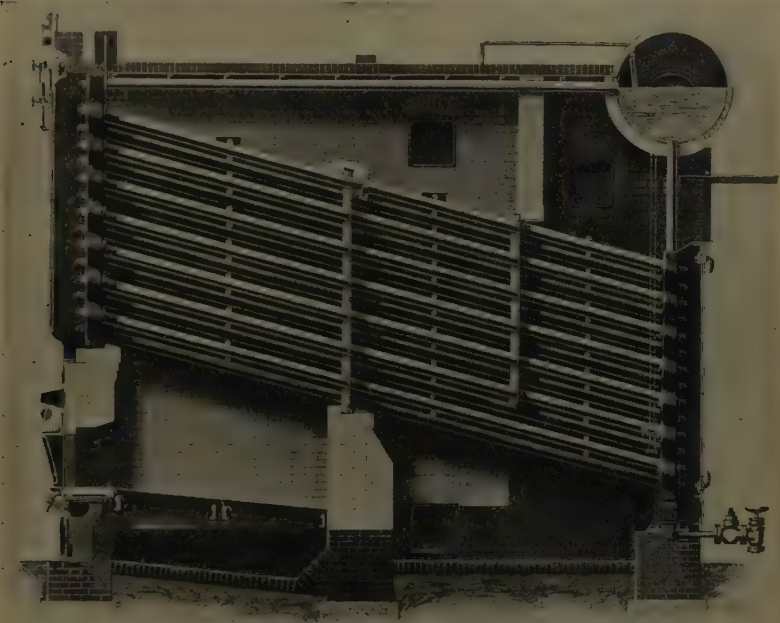
(The Babcock and Wilcox Co.)

FIG. 54.—BABCOCK AND WILCOX BOILER WITH VERTICAL HEADERS AND CHAIN-GRATE STOKER

steam-and-water drum varying from 36 to 60 inches in diameter and about 20 feet long. Large boiler units, above 300 or 400

¹ See the Babcock and Wilcox Company's book, *Steam*, for historical sketch and more detailed descriptions of various forms of their boilers.

horsepower, are built with two or more longitudinal drums side by side and spaced 3 or 4 feet apart. Below the drum or drums is a nest of straight water tubes. Each tube is usually 4 inches in diameter and 16 to 20 feet long. The nest of tubes is inclined at an angle of about 15 degrees with the horizontal. The tubes are arranged in vertical sections. The ends of the tubes in each section are expanded into sinuous cast-iron or wrought-steel headers, and



(The Babcock and Wilcox Co.)

FIG. 55.—BABCOCK AND WILCOX VERTICAL HEADER CROSS-DRUM BOILER

the stronger material being used in boilers built to withstand pressures greater than 160 pounds per square inch. The sinuous headers serve to hold the tubes in a staggered arrangement, thus causing a maximum of turbulence and mixing of the hot gases of combustion flowing over them. In Fig. 53 the headers are set in an inclined position perpendicular to the axes of the tubes. In Figs. 54 and 55 the headers are set perpendicular to the drum axis.

Each header is connected to the drum through a circulation tube, the upper end of which is expanded into a crossbox riveted to the under side of the drum. A Babcock and Wilcox superheater, consisting of a nest of hairpin-shaped tubes, as shown in Figs. 53, 54, 58, and 59, may be interposed between the drum and water tubes just above and back of the first gas passage across the tubes.



(The Babcock and Wilcox Co.)

FIG. 56.—BABCOCK AND WILCOX
HEADER

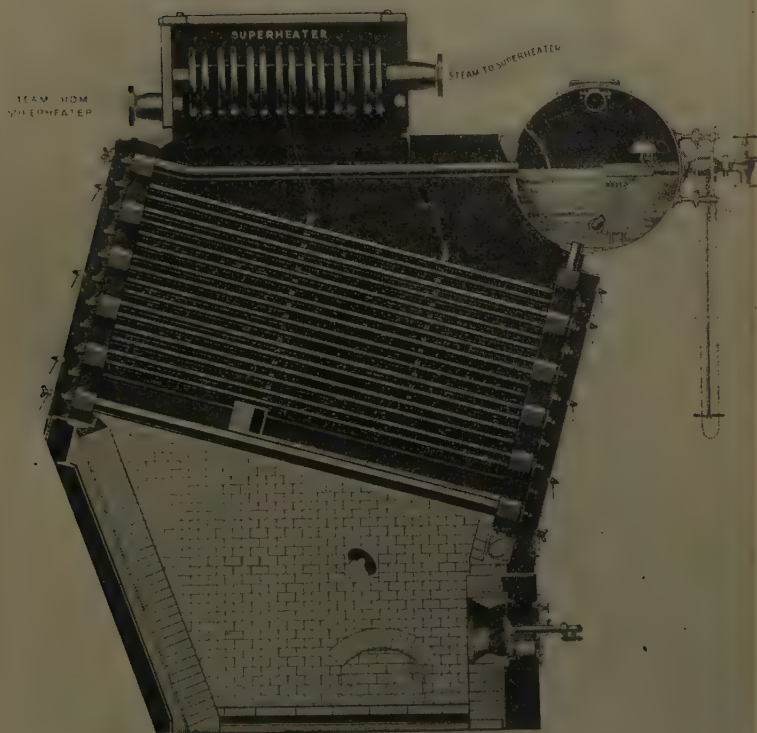
The furnace arrangement with grate bars for hand-firing of coal is shown in Figs. 53 and 55. The general arrangement of a furnace equipped with a Babcock and Wilcox chain-grate stoker is shown in Fig. 54. A marine boiler equipped to burn fuel oil is shown in Fig. 57.

Baffle walls of refractory material are placed between the water tubes, generally at two locations, to direct the hot gases of combustion across the tubes three times before they enter the flue leading to the stack. The superheater is located between the first and second passes. This is called *convection-type superheater* to distinguish it from the *radiant-heat superheater* shown in Fig. 58.

It is seen that the Babcock and Wilcox boiler is built up of sections of water tubes. The height of each section (the number

of tubes high) and the number of sections used determine the area of heating surface, and consequently the rated capacity of the boiler. Thus very large boilers may be shipped conveniently in sections to be assembled at the plant. But the special advantages of building boilers in sections are the elimination of large flat surfaces which need to be strengthened by use of stay bolts and

braces, the much reduced explosion hazard, and much higher operating pressures.



(The Babcock and Wilcox Co.)

FIG. 57.—BABCOCK AND WILCOX MARINE BOILER WITH MECHANICAL OIL BURNER

The cross-drum style of Babcock and Wilcox water-tube boiler is shown in section by Figs. 55, 57, and 59. The special advantage of the cross-drum style of boiler is that it may be installed where

head room is limited and where the weight factor is important, as on board ship. Since about 1920, the cross-drum type of boiler is being installed in many new modern central-power stations.

Fig. 58 shows longitudinal and transverse sections of a 768 hp. Babcock and Wilcox boiler equipped with Foster water-walls along the sides of the furnace, Foster radiant-heat superheater along the most exposed portion of the rear wall, and a Foster convection superheater placed in the usual location between the first and second passes above the tube bank. Pulverized fuel is prepared by an Aero Unit pulverizer. The pulverizer receives air preheated from the inclined floor of the furnace. This boiler and setting permits continuous boiler operation at loads of 250 per cent of rated capacity at high boiler efficiency.

Fig. 59 is a side sectional view of a modern design of cross-drum boiler, combining superheater, economizer, and air preheater for base-load station service. It was placed in service in November, 1926, in the Calumet Station of the Commonwealth Edison Company.

The boiler is built for a working pressure of 375 pounds per square inch. It consists of a single cross-drum 48 inches in diameter, and there are 280 $3\frac{1}{4}$ -inch tubes arranged in 40 sections, each section being 7 tubes high and 17 feet long. The economizer is of the counter-flow, loop type, and is placed in a downtake just as the gases of combustion leave the last boiler pass. The air preheater is of the single-pass counterflow tubular design placed in the uptake to the stack and just beyond the economizer. The areas of the various heating surfaces are as follows:

Boiler surface.....	5 938 sq. ft.
Furnace heating surface.....	2 460 sq. ft.
Economizer surface.....	8 365 sq. ft.

Total water heating surface..16 763 sq. ft.

Air-heater surface41 700 sq. ft.

The furnace is water-cooled and is arranged to burn Illinois coal in pulverized form. The heating value of the coal is about 12 000 B.t.u. per pound.

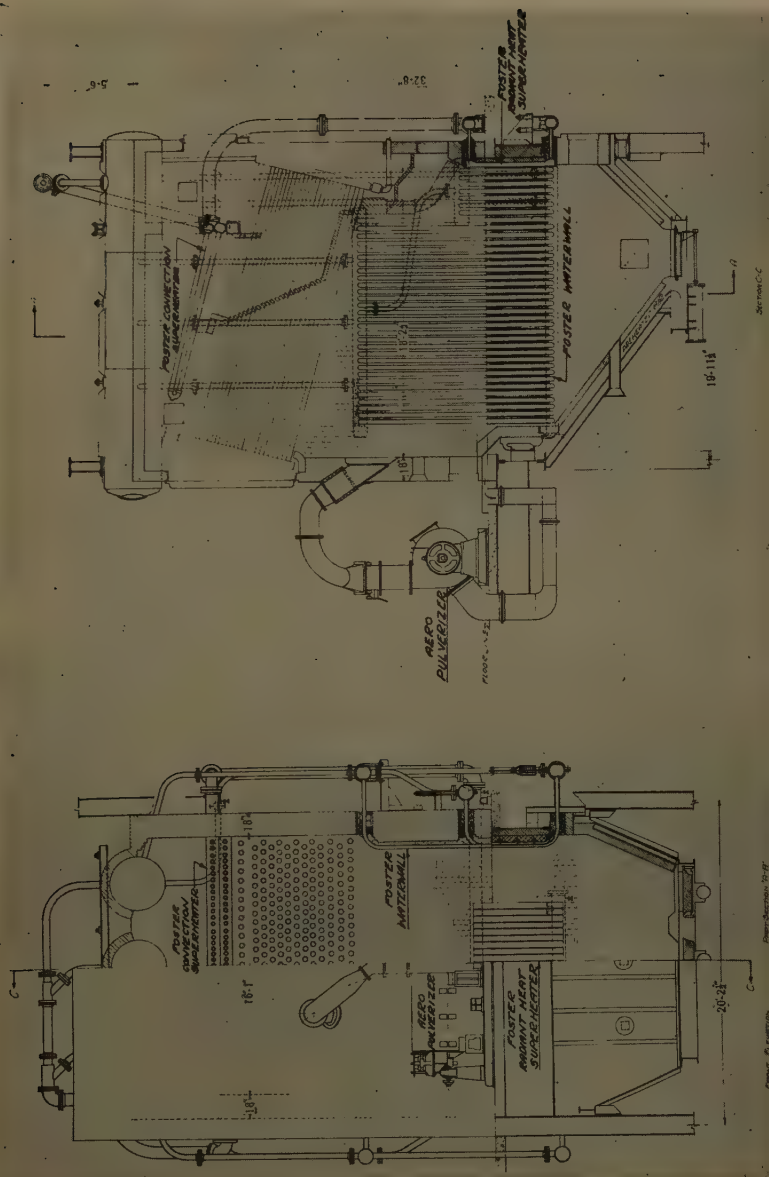
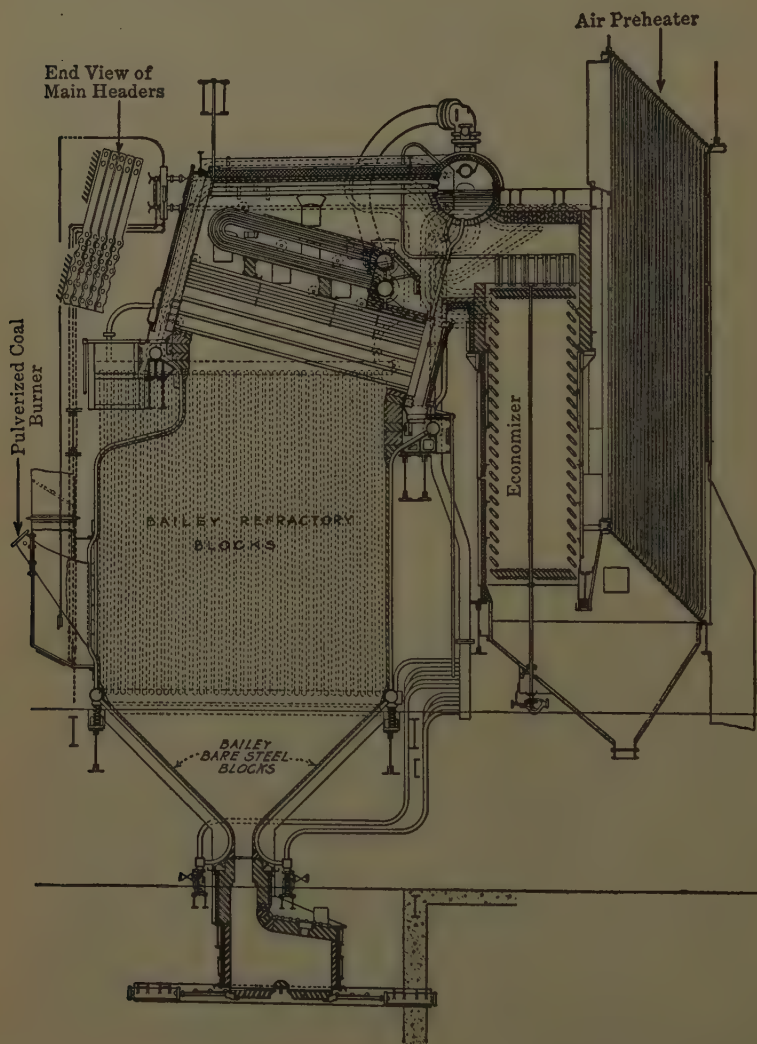


FIG. 58.—BABCOCK AND WILCOX BOILER WITH FOSTER SUPERHEATERS AND WATER WALLS

(Foster Wheeler Corporation)

The boiler is guaranteed to evaporate 200 000 pounds of water per hour and is expected to carry peak loads of 300 000 pounds. These rates reduced to a basis of pounds of evaporation per square foot of surface is shown in Table 37.



(The Babcock and Wilcox Co.)

FIG. 59.—MODERN DESIGN OF B. AND W. CROSS-DRUM BOILER AND FURNACE FOR PULVERIZED COAL

TABLE 37

CALUMET, B. AND W. BOILER PERFORMANCE RATES

TOTAL EVAPORATION, LBS. PER HR.	200 000	250 000	300 000
Evaporation, boiler, furnace and economizer, lbs. per sq. ft. per hr.	11.9	14.9	17.9
Evaporation, boiler and furnace only lbs. per sq. ft. per hr.	23.8	29.8	35.7

Since only a small portion of the economizer is generating steam, the actual evaporation rates approach more closely to the second set of figures. It will be recalled that a rated boiler horsepower is the evaporation of 3.45 pounds of water per square foot per hour. It is noteworthy that at its maximum capacity this new boiler produces steam at more than ten times normal rated capacity.¹

Modern base-load boilers are being built and operated in sizes up to 6700 rated boiler horsepower (67 000 square feet of water heating surface) and generating as high as 1 250 000 lbs. of steam per hour in a single unit. Boiler pressures up to 1400 pounds per square inch are in use. The drums of these high-pressure units are drawn from solid-steel forgings having a tensile strength of 60 000 pounds per square inch. The walls of the drum for a 1400-pounds pressure boiler are 5 inches thick. They are formed with flanged, bottle-neck ends with manhole covers bolted on in place of riveted heads.²

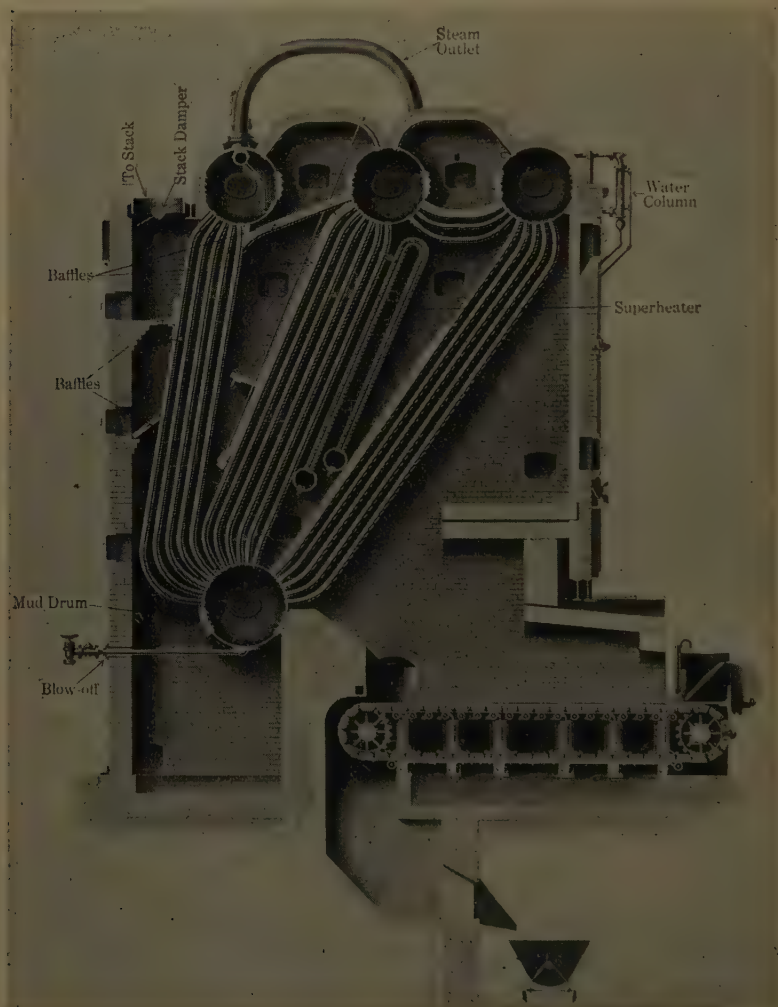
132. The Stirling Boiler. The Stirling water-tube boiler, a sectional view of which is shown in Fig. 60, first appeared in 1888. The design was then considered very radical; but it has proved itself by surviving in a highly competitive field.

The standard form consists of a series of three upper steam-and-water drums horizontally disposed at the same level and joined to a mud-drum located 16 to 20 feet below by almost

¹ For test results on this boiler see Power, Aug. 7, 1928, p. 224.

² See N.E.L.A. Prime Movers Committee Publications Nos. 289-73, June, 1929, and 289-75, July, 1929. (This material will eventually be published in the 1929 Proceedings). *Design and Tests of Large Boilers*, Transactions A.S.M.E., vol. 50 (1928), F.S.P. papers Nos. 10, 31, 32, 34, and 70.

vertical inclined water tubes. All the tubes are bent near their ends in order to enter the drums radially. The middle one of the



(The Babcock and Wilcox Co.)

FIG. 60.—STERLING BOILER AND FURNACE EQUIPPED WITH BABCOCK AND WILCOX CHAIN-GRATE STOKER

three upper drums is connected to the front drum by curved equalizing tubes above the water level which normally is at about the center line of the drums. There are also curved circulating

tubes above the water level between all three drums. The rear and middle drums only are connected by a set of curved equalizing tubes above the water level. The steam nozzle and safety valves are attached to the rear drum. The relatively cold feed water enters the rear drum and is distributed throughout its length by a trough riveted to one side. The steam leaves the drum through a dry pipe along the top similar to that shown in Fig. 64.

The tubes are usually arranged in three banks (there may be four banks with four upper drums in very large boilers) with baffle walls between so arranged that the path of the hot gases is in the main directed along the tubes. The gases flow up the front bank, which is above and slightly back of the furnace, down along the middle bank and up the rear bank. A superheater coil may be inserted between the first and second bank. The usual 3-pass arrangement is seen in Fig. 60. The mud-drum is supported in position by simply hanging suspended by the tubes. The upper three drums are supported on saddles riveted to a steel frame somewhat similar to that for the Heine boiler as shown in Fig. 65. The bent-tube type of construction makes ample provision for equalization of temperature strains incident to the generation of steam.

The drums and tubes may be readily cleaned through man-hole openings in the drum heads. The furnace setting is built inside and independent of the steel framework. The setting is made up in modified forms to suit the kind of fuel and method of firing employed.

The bent-tube style of boiler seems to be well adapted for operation in large units and high steam pressures. Units of 4000 rated boiler horsepower capacity have been in successful operation for several years, often carrying loads of 300 or even 500 per cent a large portion of the time. Double-ended, bent-tube boilers having 67 470 sq. ft. of heating surface, which upon test generated a maximum of 1 250 000 lbs. of steam per hour at a pressure of 425 lbs. and 725° F., have recently been put into service by the New York Edison Company. Boiler units of this type which can generate peak loads of 1 750 000 lbs. of steam per hour are available.

133. **The Heine Water-Tube Boiler.** The Heine boiler is manufactured in three different styles, the longitudinal-drum, the cross-drum, and the bent-tube, as illustrated in Figs. 61 to 65 inclusive. In the longitudinal-drum and cross-drum styles, straight water tubes are secured at their ends in box-like water legs having large flat surfaces which are strengthened by numerous stay bolts screwed and riveted into the two surfaces of the



(Heine Boiler Co.)

FIG. 61.—HEINE FOUR-PASS WATER-TUBE BOILER

water legs. The longitudinal drum is set with its axis parallel to the tubes. When erected, the entire boiler is set with the front end highest so that the drum and tube-axes slope upward toward the front. This position aids in maintaining a vigorous water and steam circulation whose direction is up the front water leg, down the back leg, and toward the front through the tubes.

The drums for all three boiler styles are made in diameters varying from 36 to 60 inches depending upon the size of the boiler. In the longitudinal-drum style the drum is from 18 to 22 feet long. The cross drums may be shorter or longer depending upon the size

of the unit. The tubes are generally 4 inches in diameter and 16 to 20 feet long.

In the longitudinal and cross-drum styles, the paths of the hot gases may be directed, by suitable baffles, either across the tubes

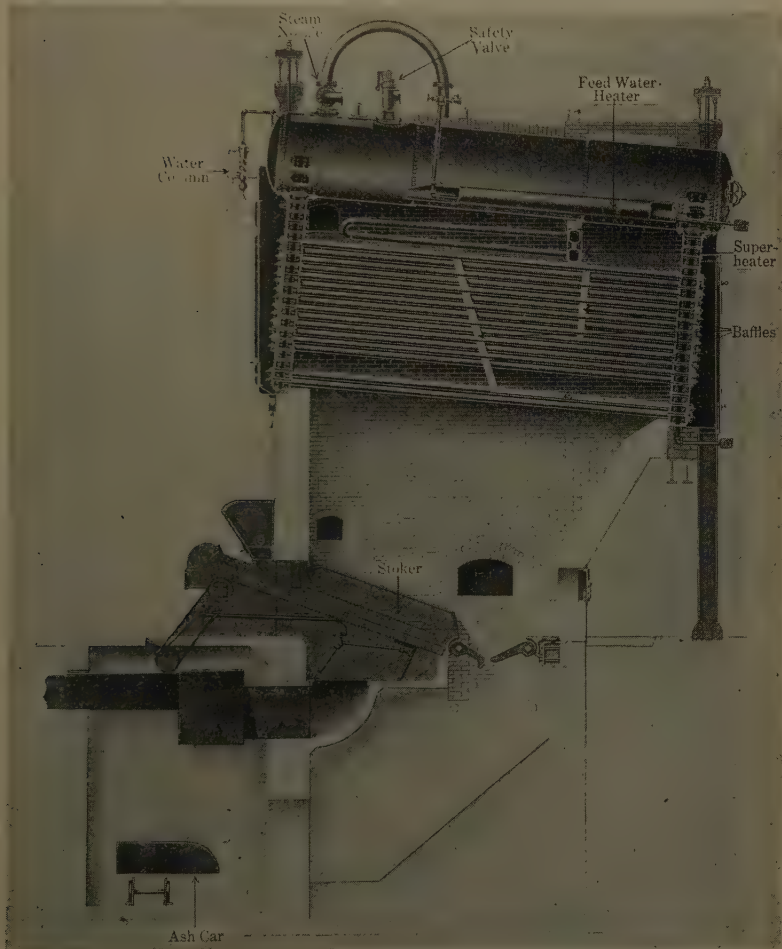
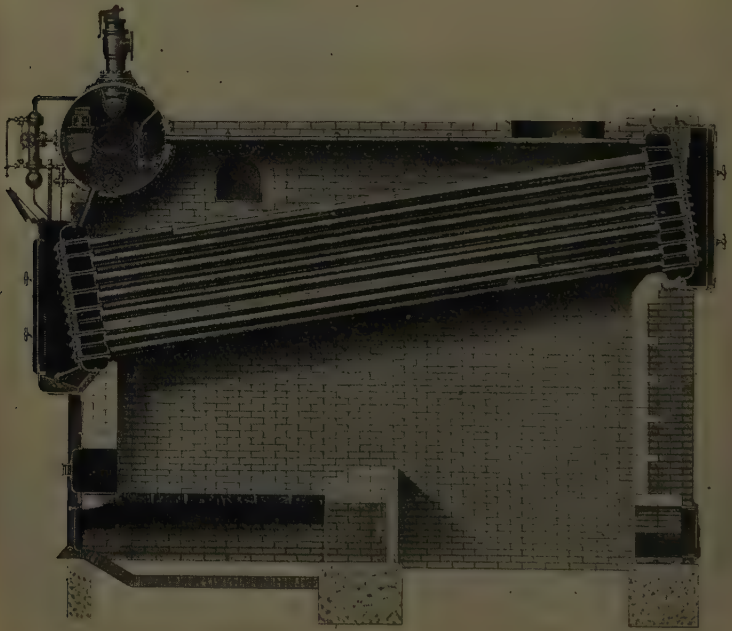


FIG. 62.—HEINE THREE-PASS WATER-TUBE BOILER WITH SUPERHEATER AND WESTINGHOUSE UNDERFEED STOKER

as in Figs. 61 and 62 or parallel to the tubes as shown in Figs. 63 and 65. The cross-baffling of Fig. 62 seems to be the more efficient arrangement and is more frequently used in straight-

tube boilers. Superheating steam coils may be placed above the tube bank of either longitudinal or cross-drum boiler as shown in Fig. 62.

In the longitudinal-drum boiler, a dry pan with many small holes in the sides and bottom is attached inside the drum just below the steam-outlet nozzle which is located at the front end of the drum. A deflecting plate which aids in directing and



(Heine Boiler Co.)

FIG. 63.—HEINE SINGLE-PASS CROSS-DRUM BOILER

maintaining water circulation is riveted to the front end of the drum just above the water leg. The feed water enters an internal heater and distributor which is located on the bottom of the drum near the rear end. These features may be seen in Fig. 62.

The cross-drum boiler is equipped with a dry pipe which is located in the top of the drum and connected to the steam-outlet nozzle. The feed water enters a heating and distributing trough

which is riveted to the side of the drum. (See Fig. 64.)

In Fig. 65 a view of the V-type Heine boiler is shown. Its chief features are the bent tubes, set nearly vertical, and arranged in banks. A series of cross drums, three steam and water drums at the top, are placed on the same level and one mud-drum below, the drums being connected by the bent water-tubes. Different arrangements of the tube banks with suitable baffles provide for 3, 4, or even 5 gas passes. The general direction of flow of the hot gases is parallel to the tubes. In some very large boiler

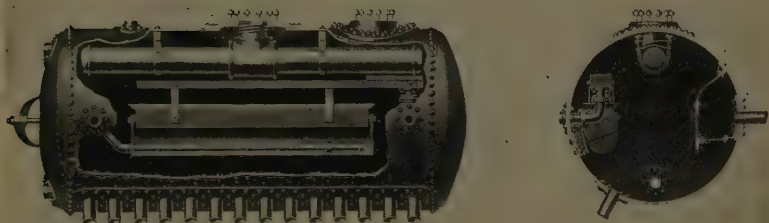
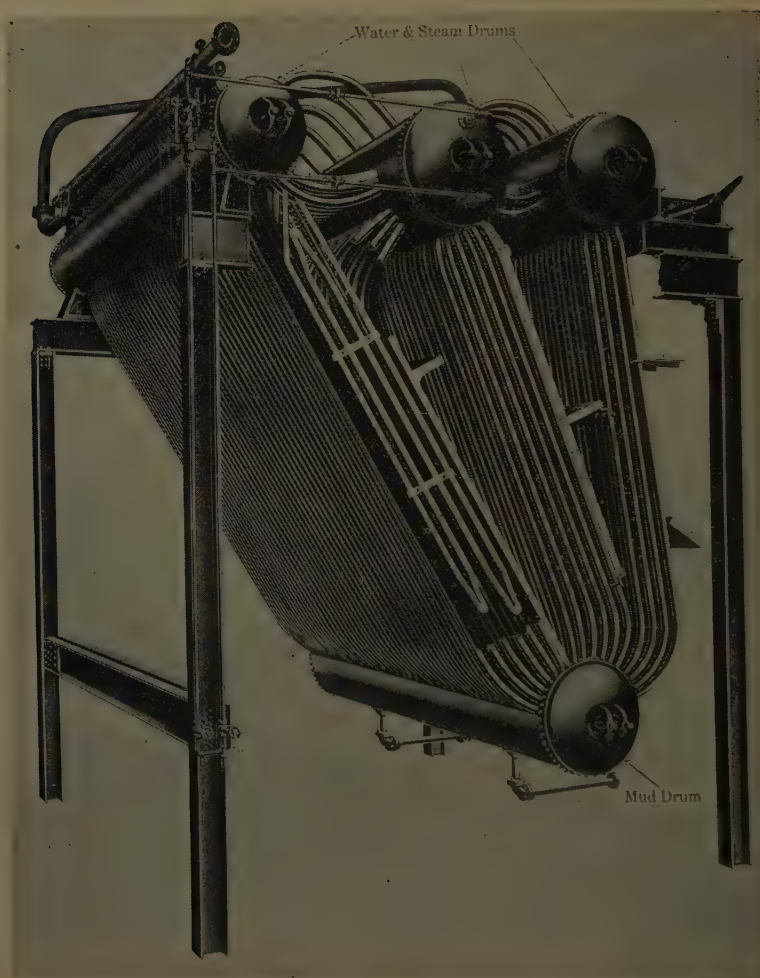


FIG. 64.—SECTION OF HEINE CROSS-DRUM SHOWING DRY PIPE AND FEED-WATER INLET

units four steam-and-water drums are employed with two mud drums below. The boiler is supported on a steel framework independently of the masonry setting, as illustrated in Fig. 65. In many respects, the V-type Heine boiler is very similar to the Stirling boiler illustrated in Fig. 60. The chief difference seems to be that water-circulating tubes below the water level are provided between all the upper drums of the Heine boiler, while the water-circulating tubes are placed only between the front and middle drums of the Stirling boiler.

The Heine Boiler Company announce that they regularly build V-type boilers in rated sizes varying from 250 to 3800 boiler horsepower and for steam pressures up to 325 pounds per square inch.



(Heine Boiler Co.)

FIG. 65.—HEINE V-TYPE THREE-PASS BOILER AND SUSPENSION FRAME

134. Boiler Furnaces.¹ The function of a boiler furnace is to furnish conditions favorable for carrying on continuously a rapid rate of combustion, thus liberating large quantities of heat energy at high temperatures, for heat flow into the boiler is proportional to nearly the square of the temperature difference

¹ See R. T. Haslam and R. P. Russell, *Fuels and Their Combustion*, 1926, pp. 333-508.

between the hot gases of combustion and the water within the boiler. Combustion must be supported by an adequate supply of air, properly distributed. There must not be a large excess of air for it tends to decrease the furnace temperature and increase the weight of stack gases, dissipating increased amounts of heat energy into the atmosphere. There must be sufficient space in the furnace so arranged that every molecule of burning fuel is able to meet and combine with the necessary molecules of oxygen.

For the purposes of studying furnace performance, the air supply is divided into primary and secondary air. The primary air enters the furnace from beneath the grate bars in the case of solid fuels, or it enters checkerwork openings in the floor of the furnace in case of liquid fuels. The secondary air enters the furnace above the fuel bed. Thus there may be listed six important factors which enter the problem of rapid and efficient combustion. They are: the fuel, primary air, secondary air, high temperature of combustion, turbulence of the gases, and furnace volume. Turbulence is a function of velocity and furnace form. Gas velocities in modern furnaces vary from about 5 feet per second to 30 feet per second.

The type of furnace and its construction details depend largely upon the kind of fuel and its state when it enters the furnace. The capacity of a furnace depends first upon an adequate supply of primary air. The efficiency of furnace operations depends upon the form and volume of furnace and upon the amount, dispersion, and turbulence of the secondary air. Efficient transfer of heat through the boiler surfaces at high rates per unit of boiler area depends primarily upon a large temperature difference, and secondarily upon high gas velocities and rapid water circulation. Excess of primary and secondary air materially decreases the furnace temperature and increases the heat loss up the stack. Complete combustion can be realized only when the temperature of the mingling gases remains above the kindling point. This requires a large furnace volume so disposed that the hot gases do not come into contact with the relatively cold surface of the boiler until combustion is complete. The proper combination of all these factors to give high efficiency constitutes a separate problem

for every kind of fuel. Special furnace designs are required for high and low volatile coals, for fuel oil, for gaseous fuel, and for hand and stoker firing.¹

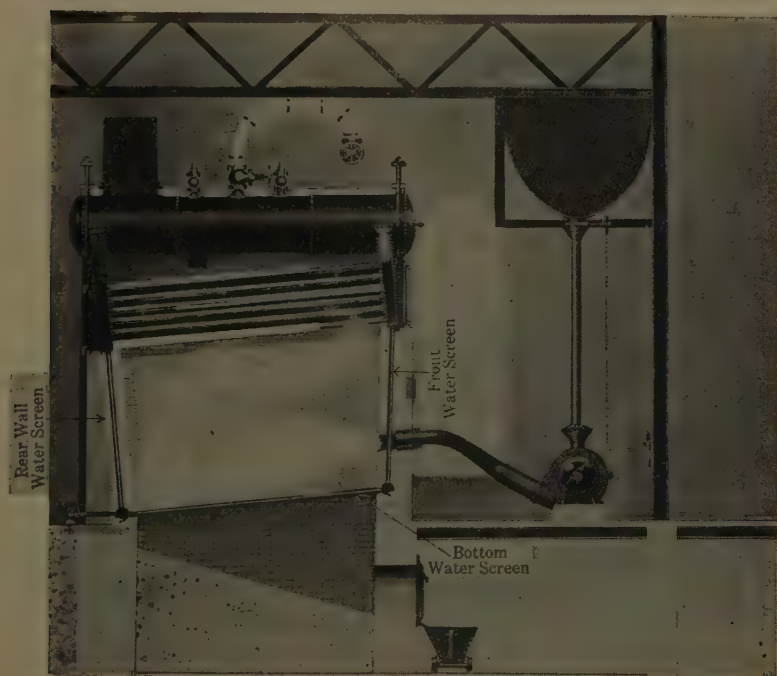
The tendency in modern central power stations is toward increased use of water-cooled walls, radiant-heat superheaters, air preheaters (heating the air to 400° F. or 500° F.) and pulverized coal with turbulent burners. These features play an important part in making possible larger units, higher rates of driving, higher efficiencies, and lower power costs.

The furnaces shown in Figs. 51, 53, and 55 represent the older designs. The vertical distance from the grates to the boiler shell or the water tubes is too short for burning highly volatile bituminous coals smokelessly and efficiently. The vertical distance from the grate bars to the boiler shell in Fig. 51 should be from 3 feet to 5 feet, depending upon the size of the boiler and the size of the overloads which it is expected to carry. In Fig. 53, the height just inside the fire door should be approximately 6 feet. Well-arranged furnaces for hand-firing of highly volatile coals are shown in Figs. 52 and 63. Suitable furnaces for stokers are shown in Figs. 54, 60, and 62.

Oil-fired furnaces are shown in Figs. 57 and 79. Figs. 58, 59, and 66 show furnaces arranged for burning pulverized coals. The side walls and furnace floor are provided with water screens consisting of a series of water tubes spaced from 9 to 15 inches apart and forming a part of the water-circulation system. The furnace temperature produced by pulverized coal is very high, often above the melting temperature of the firebrick lining, and nearly always above the fusing point of the ash. Such high temperatures burn out the furnace linings rapidly, and molten ash flowing down the walls erodes the firebrick seriously. If the ash reaches the bottom of the furnace in the molten state, it forms, upon cooling, a solid mass of clinker which is very difficult to remove. By the use of properly arranged water-cooled walls (Figs. 59 and 60) or water screens (Fig. 66), these expensive troubles are largely prevented.

¹ See Proceedings of the National Electric Light Association, 1924, pp. 1306-1350, and 1927, p. 1012-1086, and pp. 1240-1358.

In Fig. 67 is shown a sectional view of an Aero Unit coal pulverizer whose development began in 1898. The Aero Unit is a multistage pulverizer provided for removal of the pulverized coal by air currents moving at controlled velocities. The machine is driven by a direct-connected electric motor which is not shown. The coal, crushed to a suitable size (passing a 1-inch or an $1\frac{1}{4}$ -inch



(Erie City Iron Works)

FIG. 66.—WATER-TUBE BOILER WITH SEYMOUR WATER-COOLED FURNACE FOR PULVERIZED COAL

screen), enters the left end of the pulverizer through an adjustable feed mechanism and falls into the first stage of the pulverizer by gravity. The coal is reduced by impact with the revolving impellers and is thrown by centrifugal force against the manganese-steel liners in the shell. Air, preferably preheated, admitted at the coal-feed mechanism carries the particles of coal over the first diaphragm into the succeeding stage only after they have been reduced to a size which the air velocity is able to move in opposi-

tion to the centrifugal force. In the succeeding stages this process is repeated. Here the pulverizing action is a combination of abrasion of particle on particle and particle against impeller-blade or liner. The velocity of the air controls the fineness. The operator controls the air velocity by adjusting dampers at the inlet and exit of the pulverizer.

The fineness of the coal conveyed to the furnace is independent of changing clearances between the revolving and stationary parts due to wear, and it is independent of size or character of coal supplied to the pulverizer. It is dependent only upon the air velocity



(Foster Wheeler Corporation)

FIG. 67.—AERO UNIT COAL PULVERIZER

as controlled by the operators. Any degree of fineness desired can be obtained even up to 90 per cent of the coal pulverized to pass through a 200-mesh screen and 99 per cent through a 100-mesh screen. But the best results are obtained when the coal is sufficiently fine to permit complete combustion while suspended in the furnace atmosphere. In modern furnaces, this result can be realized if 99 per cent of the coal will pass through a 40-mesh and 80 per cent through a 200-mesh screen. It is a waste of power to pulverize the coal finer than required to meet the specification stated above. In the unit system, the pulverized

coal is conveyed directly to the boiler furnace as fast as pulverized. The unit system is in successful use with boiler units as small as 250 rated boiler horsepower.¹

135. Furnace Volume. The volume of the combustion space provided in the furnace per B.t.u. developed, or per square foot of boiler heating surface, is an important factor which has been investigated as it deserves only in recent years. Practice at present is far from standardized. The volume per square foot of heating surface for stokers has been increasing. The furnace volume for burning pulverized coal is so large for the very large boiler units now being used that the initial and operating furnace costs are excessive. Intensive research is being carried on to develop methods of burning pulverized coal efficiently in smaller furnaces. It is believed that turbulence of the burning gases is an important factor in increasing combustion rates per unit of furnace volume.¹ A few representative values are given in Tables 38 and 39.

TABLE 38²
BOILER AND FURNACE DATA FOR PULVERIZED COAL

STATION	BOILER SURFACE Sq. Ft.	STEAM GENER- ATED LBS. PER HR.	SPECIFIC FURNACE VOLUME CU. FT. PER Sq. Ft.	B.T.U. LIBER- ATED PER HR. PER CU. FT.	EXCESS AIR PER CENT	TYPE OF BURNER
Merrimac Chem....	3 760	16 600	0.467	14 000	40	Flare
Merrimac Chem....	3 430	20 700	0.802	17 100	40	Turbulent
Bayonne.....	2 755	13 706	0.093	75 060	38	Turbulent
Calumet No. 22....	17 288	285 000	0.715	30 000	25	Calumet
Buffalo No. 13....	12 515	225 000	0.406	60 000	20	Calumet
Columbia.....	15 110	160 000	0.795	18 500	39	Vertical
Lakeshore.....	30 600	286 200	0.852	15 000	45	Lopulco
East River.....	67 470	800 000	0.580	25 000	—	Lopulco

Recently, the East River boilers of Table 38 generated, at peak load conditions, about 39 000 B.t.u. per cu. ft. of furnace volume.

¹ See W. H. Brooks in Transactions A.S.M.E., vol. 47 (1925), p. 101.

² Transactions, A.S.M.E., vol. 51 (1929), F.S.P.-Paper No. 28.

TABLE 39¹

BOILER AND FURNACE DATA FOR STOKERS, OIL AND GAS BURNERS

STATION	BOILER SURFACE Sq. Ft.	STEAM GENER- ATED LBS. PER HR.	SPECIFIC FURNACE VOLUME CU. FT. PER Sq. Ft.	B.T.U. LIBER- ATED PER HR. PER CU. FT.	EXCESS AIR PER CENT	TYPE OF STOKER OR BURNER
Crawford Ave.	16 220	157 900	0.344	38 000	34	Chain grate
Twin Branch.	14 086	166 000	0.387	45 700	28	Chain grate
Richmond.	15 700	224 000	0.427	31 000	22	Underfeed
Saxton No. 5.	11 120	138 000	0.426	41 000	24	Underfeed
Long Beach No. 3. . .	34 164	350 000	0.580	21 000	—	{ Combined gas and oil burner
Great Western Power	22 250	300 000	0.780	23 000	—	

¹ See Transactions A.S.M.E., vol. 50 (1928), F.S.P. papers Nos. 70, 71, 72 and 78. N.E.L.A. Publication No. 289-75, July, 1929.

Table 40 gives the range of combustion rates which has been found possible in various kinds of furnaces.

TABLE 40
FURNACE VOLUME AND B.T.U.

FUEL-BURNING SYSTEM	B.T.U. PER HOUR PER CU. FT. OF FURNACE VOLUME
Hand-fired furnaces	10 000 to 20 000
Pulverized coal	12 000 to 200 000
Chain grate stokers	15 000 to 50 000
Underfeed stokers	20 000 to 160 000
Locomotives	65 000 to 75 000
Fuel oil, steam atomizing	40 000 to 90 000
Scotch marine, hand firing	40 000 to 150 000
Fuel oil, mechanical atomizing	20 000 to 180 000
Gas, surface combustion *	500 000 to 700 000

* See R. T. Haslam and R. P. Russell, *Fuels and Their Combustion*, 1926, p. 308-309.

136. Automatic Stokers.¹ In 1785, James Watt developed and put into practice ideas to control the distillation of the volatile matter of coal and to properly burn it. This constituted

¹ See R. T. Haslam and R. P. Russell, *Fuels and Their Combustion*, 1926, pp. 383-425.

Worker and Peebles, *Mechanical Stokers*, 1922. Stoker Manufacturers' Assn., Condensed Catalog of Mechanical Stokers, 1924.

Two hundred and thirty-seven years of stoker development. Combustion, vols. 8 and 9. 1922-1923.

Proceedings National Electric Association, 1925-1928 incl.

one of the earliest attempts to develop a mechanical stoker, but it was not automatic. Stokers have been in use in European countries since about 1870. They did not come into general use in the United States until about 1900.

By the use of the highly developed modern automatic stokers, coal may be fed into the furnace continuously at rates which may be varied either manually or automatically to meet the rapidly varying load demands. Progressive distillation and efficient

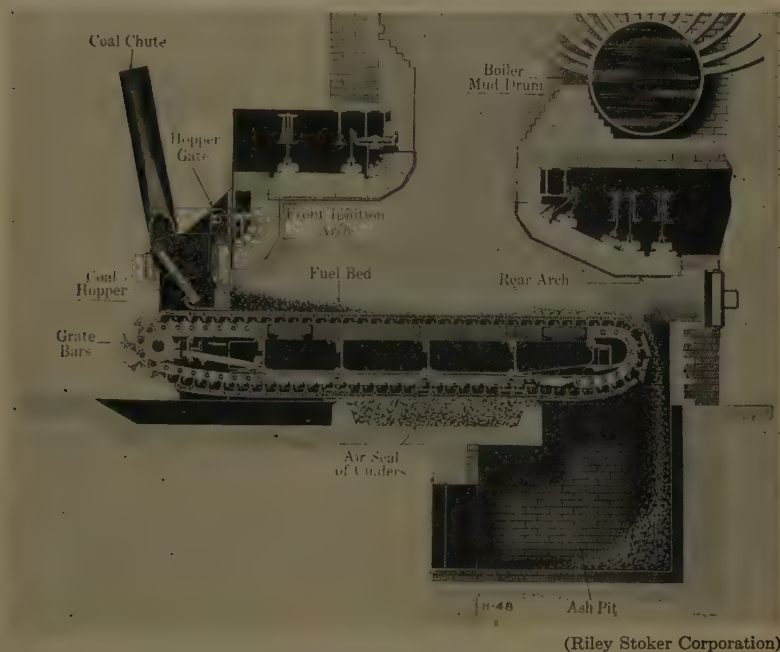


FIG. 68.—HARRINGTON CHAIN-GRATE STOKER

combustion of the volatile matter are readily maintained. Continuous automatic removal of the ash and cinders is accomplished. The air supply may also be regulated automatically, and its proper distribution secured for the highest possible efficiency. Appreciable savings in labor costs are possible. One man is able to attend to the operation of stokers for 5000 to 10 000 boiler horsepower. Stoking by hand, one man can handle about 500 boiler horsepower. Stokers may profitably be installed in boiler

units of 200 horsepower and larger. In a few instances they have been installed under boilers of 100 horsepower. Their installation under boilers smaller than this is not usually profitable because their first cost, operating, and maintenance charges more than balance the fuel savings. The advisability of installing stokers under small boilers can be determined only by a study of the various factors involved in each case.

Automatic stokers naturally divide themselves into three groups, namely, chain-grate, underfeed, and overfeed stokers. All of the types have reached a high degree of perfection and will burn all kinds of coal fairly well. But each type of stoker is most highly successful with certain ranks and grades of coal.

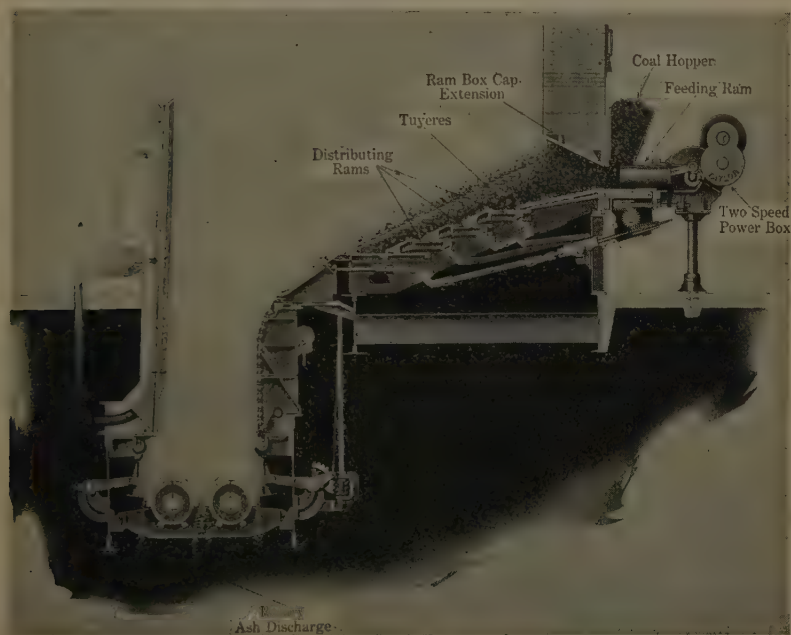


(Riley Stoker Corporation)

FIG. 69.—SECTION OF RILEY UNDERFEED STOKER

Chain-grate stokers, sometimes designated as traveling grates, consist of an endless chain or conveyor to which are clamped or bolted many short bars or blocks. The entire conveyor travels over sprockets or drums at the front and rear of the furnace. The frame supporting the conveyor, sprockets, and driving mechanism may be mounted on wheels and track, thus permitting the stoker to be rolled out of the furnace for inspection and repairs. Views of chain-grate stokers are shown in Figs. 54, 60, and 68.

Coal, broken to suitable size, is fed from the storage bin through chutes to the hopper at the front of the grate. A gate in the hopper regulates the depth of coal on the grate. As the fuel enters the furnace, it is ignited near the front of the furnace by an ignition arch which reflects the heat from the hot part of the fire to the point where the coal enters. Combustion progresses as the coal is carried toward the back of the furnace and is completed as



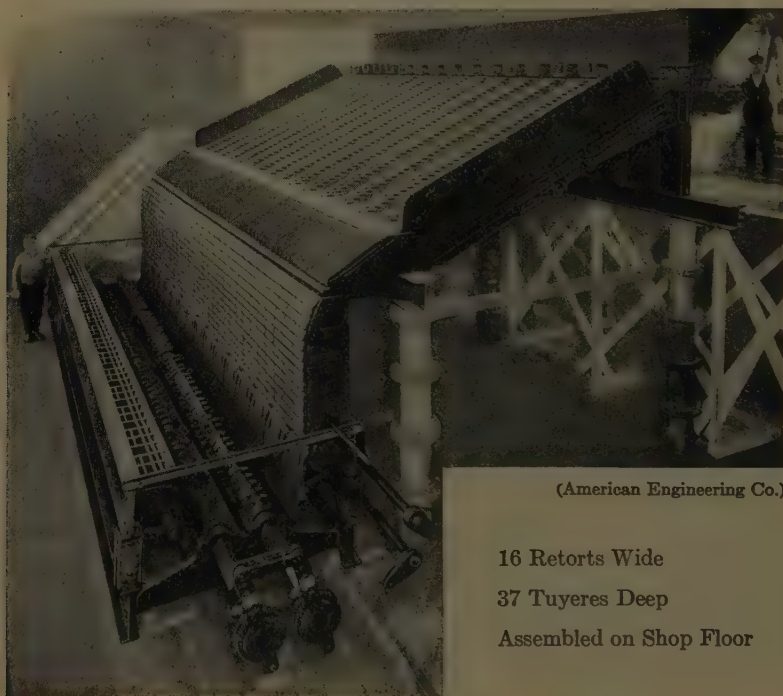
(American Engineering Co.)

FIG. 70.—SECTION THROUGH TAYLER STOKER AND FURNACE

the chain passes over the rear drum. The ash and clinker are dropped into the ash pit under the rear end of the stoker.

Underfeed stokers are set either horizontal or inclined downward toward the rear of the furnace. The fuel is generally fed from the front and from underneath by means of intermittent fuel rams or a continuously rotating screw conveyor. Three different makes of this type of stoker are shown in Figs 69, 70, 71, and 72. With the underfeed stoker, the raw coal is gradually coked by heat conducted down through the fuel bed as the coal

is pushed up into the burning zone by the feeding and distributing rams. The volatile matter is gradually driven off and burned as it is forced by the air blast up through the incandescent coke on top of the fuel bed. When combustion is complete, the ash is deposited on side or back dump plates or passed through rear crusher rolls depending upon the stoker design.



(American Engineering Co.)

16 Retorts Wide

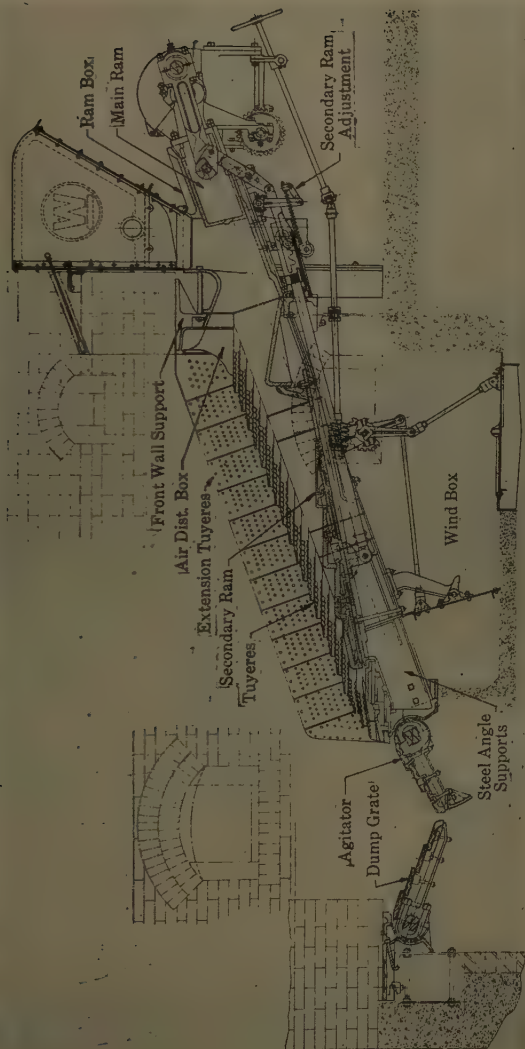
37 Tuyeres Deep

Assembled on Shop Floor

FIG. 71.—TAYLOR UNDERFEED STOKER

The rate of fuel consumption is controlled by varying the length and number of stokes of the rams per hour. The best designs are so arranged that each feeding and distributing ram may be controlled independently. In conjunction with variable rates of coal feed, the air supply which is always by forced draft must also be varied.

Two styles of *overfeed stokers* are represented in Figs. 73 and 74. In the first, the coal is fed at the front of the furnace, the grate bars being disposed transversely and in an incline



(Westinghouse Electric & Manufacturing Co.)

FIG. 72.—WESTINGHOUSE UNDERFEED STOKER

extending downward toward the rear of the furnace. The coal is propelled backward and downward by a rocking movement of the grate bars as the combustion proceeds. The ashes are dumped periodically by hand or by aid of steam-actuated levers. During the dumping process a guard grate is automatically raised to prevent the loss of unburned coal. The rate of feeding coal and the motion of the grate bars are controlled independently by controlling the speeds of the driving motors. The air is generally supplied to overfeed stokers by means of natural draft.

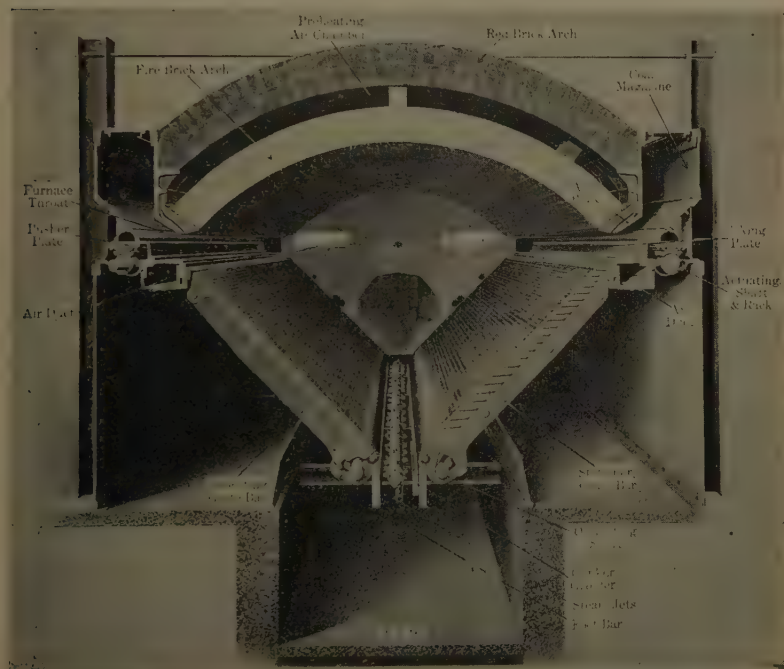


(Westinghouse Electric & Manufacturing Co.)

FIG. 73.—WESTINGHOUSE—RONEY OVERFEED STOKER

In the second style of overfeed stokers, Fig. 74, the coal is fed from magazines arranged along the entire length of the furnace on both sides. The grate bars are arranged in two inclined planes which meet at the center, forming a V-shaped furnace. The longitudinal axes of the grate bars extend up and down the incline. Alternate bars are stationary and the others are oscillated by a rocker shaft on each side, usually placed at the bottom of the incline. The coal is carried from the magazine into the furnace

by a pusher plate actuated by rocker shaft and rack. The rate of feeding is controlled by regulating the arc of travel of the rocker shaft. The movement of the oscillating grate bars breaks up the fuel bed and keeps it moving down the incline at such a rate that only the ash and clinker remain at the bottom. The Murphy stoker and furnace shown in Fig. 74 belong in this class. This furnace is capable of handling any kind of coal and is par-



(Riley Stoker Corporation)

FIG. 74.—MURPHY OVERFEED STOKER

ticularly well suited for small or medium-sized boilers. It is claimed that they may be used economically for boilers as small as 50 rated horsepower. The firebrick arch is required in all sizes of the Murphy stoker in order to eliminate smoke.

Chain-grate stokers are especially well adapted to handle non-coking coals which do not require agitation in order to burn them efficiently. It is the only stoker suitable for small sizes of anthracite and coke breeze. Coals with ash of low fusing temper-

ature may be readily handled; because the fuel bed is unagitated, the ash remains in an oxidizing atmosphere which is not favorable for clinker formation. Therefore non-coking coals of high ash content are burned efficiently. Coals of the West and Middle West mostly belong in this class. Mixed coals cannot be handled efficiently, because the time during which the coal must be completely burned is determined by the length and rate of travel of the grates. When the fuel and ash reach the end of the chain they are dumped into the ash pit. A highly volatile coal, like bituminous or subbituminous, ignites at a lower temperature and burns more rapidly than does a low volatile coal such as anthracite or coke breeze. If the rate of grate travel is adjusted to suit the requirements of the quick-burning bituminous coal, some of the slow-burning coal will pass into the ash pit unburned, producing a high ash-pit loss. If the length of grate and rate of travel is adjusted to suit the slow-burning coal, the more volatile coal is burned out before the grate arrives at the back of the furnace, thus leaving air holes in the fuel bed through which large excesses of air enter. This causes a high stack loss. Other inherent defects are: as the fuel approaches the rear end, there is no means of crowding it together to maintain a fuel bed of uniform thickness, thus large quantities of excess air enter the back of the furnace. It possesses poor banking characteristics.

Overfeed stokers are best adapted to burn low-ash, high-grade coals, particularly the semibituminous variety. The tendency to avalanch, high percentages of combustable in the ash under overload conditions, limited overload capacity, high-draft pressure required, and necessity for ignition arch are factors which prevent a wide adoption of this type of stoker especially in the larger power stations.

Multiple-retort *underfeed stokers* are capable of burning efficiently practically all types of bituminous coals. Underfeed stokers do not require an ignition arch, which is the most costly part of the furnace to maintain. Forced draft must be employed with this type of stoker. The top of the fuel bed is at a red or white heat; hence little or no smoke and carbon dust escape to be deposited on the tubes. A large percentage of the boiler heating

surfaces looks down upon the red-hot fuel; hence much heat energy is transferred to the boiler by radiation, which is the most efficient mode of heat transmission. These stokers are capable of efficient operation at high rates of combustion. They are much used where the boilers are to be driven at 300 to 500 per cent of normal rating.

Since the arrival of pulverized coal on a large scale, the stoker manufacturers have made radical improvements in stoker design and operating characteristics in order to meet the capacity and efficiency records established by pulverized coal equipment. The most economical coal-burning system for a given plant can be selected only after all the factors involved have been studied by experienced combustion engineers.¹

137. Fuel-Oil Burners.² In order to burn fuel oil smokelessly and efficiently in any furnace, it must first be broken up into a spray of very fine droplets. This is necessary in order that every molecule of oil may meet and combine with the necessary oxygen. Atomization of oil may be effectively accomplished in either of two ways, blowing it with steam or air, or by utilizing the pressure energy of the oil itself to break it up. The older method is atomization by use of steam or air at pressures varying from 5 pounds to 100 pounds per square inch, depending upon the characteristics of the oil and the type of burner. The oil is heated to a temperature of about 150° F. before it enters the burner.

One of the successful and well-known steam-atomizing burners is shown in Fig. 75. Because of its simplicity, reliability, and effectiveness, the steam-atomizing style of burner is much used in locomotives and in small power stations. In the better forms, it is fairly economical in the use of steam, requiring from 1 to 3 per cent of the total steam generated by a boiler to operate

¹ See Transactions, A.S.M.E. vol. 50, 1928, FSP. papers Nos. 15, 65, and 70; and vol. 51, 1929, FSP. paper No. 20.

Proceedings N.E.L.A., Prime Movers Committee Reports for 1927 and 1928.

² See G. F. Gebhardt, *Steam Power-Plant Engineering*, 1925, pp. 252-260. Sixth edition.

Robert Sibley and C. H. Delany, *Fuel Oil and Steam Engineering*, 1921.

R. T. Haslam and R. P. Russell, *Fuels and Their Combustion*, 1926, p. 468.

Transactions, A.S.M.E. vol. 50, 1928, FSP. paper 13, p. 103.

its burners. The steam pressure required at the burner tip for effective atomization varies from 30 to 60 pounds per square inch gage.

Compressed air is generally used for the atomizing agent in metallurgical furnaces because of the higher flame temperatures possible and because in many cases a higher-grade product results.

The *Best oil burner* is known as an outside-mix burner, since the steam and oil do not mingle until they leave the burner. The method of operation is clearly shown in Fig. 75 (a). The



FIG. 75.—BEST HIGH-PRESSURE OIL BURNER

usual arrangement of an oil-fired furnace under a Babcock and Wilcox Marine boiler is shown in Fig. 57. The details of the furnace setting are varied to suit the type of service for which the boiler is intended.

In the smaller plants, the operation of the burner is controlled by hand. But for large boiler units in central power stations, it is more economical of fuel and labor to employ one of several available automatic-control systems which regulate the stack dampers, the atomizing steam, and flow of oil to the burners simultaneously. The control device is usually actuated by small variations (1 to 5 pounds) in the boiler pressure.

In marine plants, mechanical-atomizing burners are used almost exclusively because the loss of 2 or 3 per cent of the steam up the stack must be replaced by evaporating salt water in special equipment. This involves considerable expense, most of which is eliminated by use of the mechanical burners. In Figs.

76 and 77 are shown views of one of the few successful burners of this type.

The principle of the *Peabody-Fisher mechanical oil burner* is illustrated in Fig. 76. The oil under a pressure of about 200 pounds per square inch and at a temperature of about 250° F. enters a whirlpool chamber in such a manner as to acquire a vortical motion as it approaches the orifice. In passing through

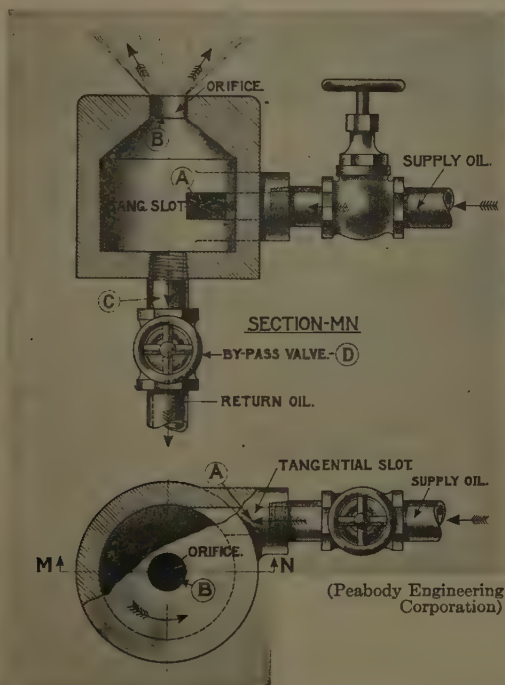


FIG. 76.—DIAGRAM SHOWING METHOD OF CONTROL OF PEABODY-FISHER OIL BURNER

the orifice, the combined effect of the pressure and the vortical velocity is to break the oil up into a very fine spray in which condition it burns smokelessly if the air supply is adequate and properly distributed. By controlling a by-pass valve attached to the whirlpool chambers, much or little of the oil may be returned to the pump intake. In this way the rate of burning oil may be varied through an exceptionally wide range without interfering

with the effectiveness of atomization because the vortical velocity and pressure are not reduced at low rates of burning. As little as 10 per cent of full burner capacity is possible with perfect combustion. It is customary to install one burner for each 200 or 300 horsepower of boiler capacity. The mechanical-atomizing type of burner is being rapidly introduced into land plants since it is more economical to operate than the steam-atomizing type.

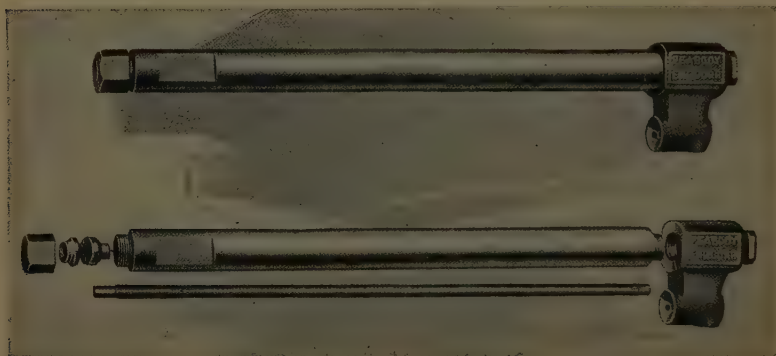
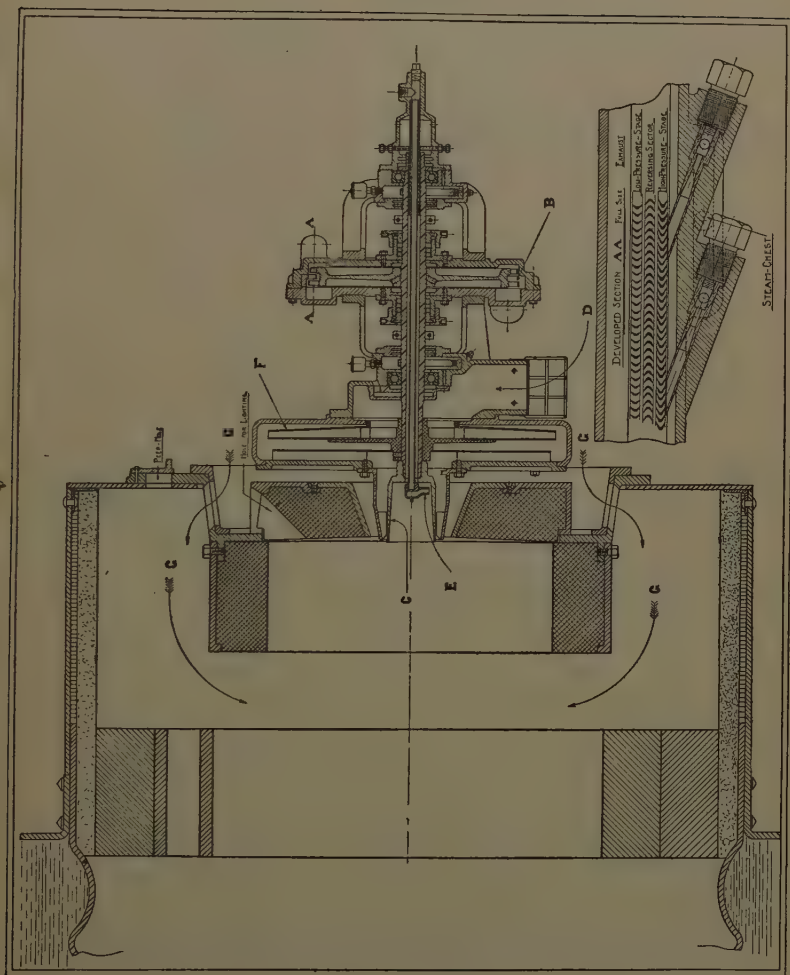


FIG. 77.—PEABODY-FISHER MECHANICAL OIL BURNER

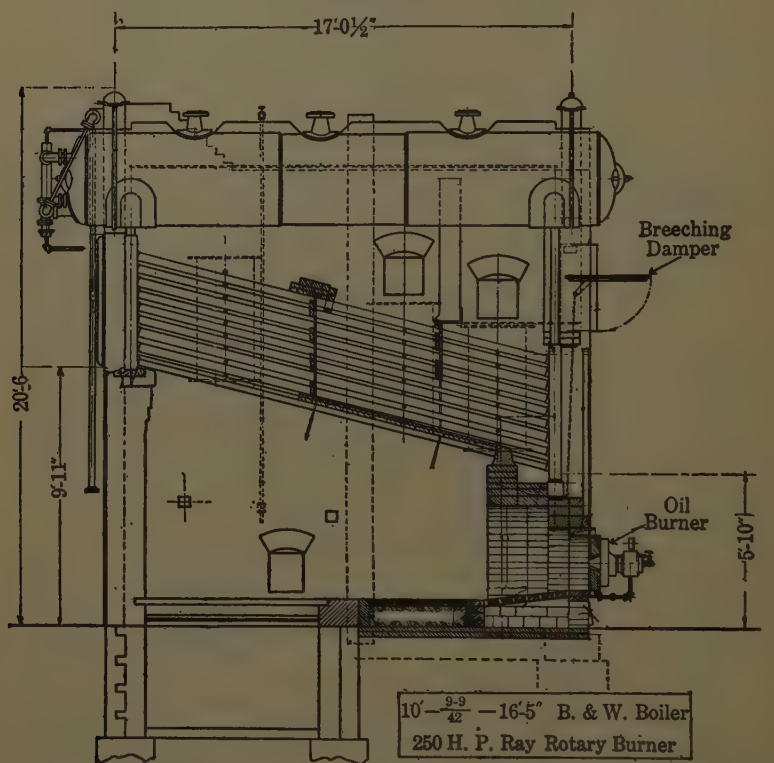
The Ray rotary fuel-oil burner, Fig. 78, differs radically from most mechanical burners in that it does not utilize high pressure to atomize the oil, but employs centrifugal force. The oil drools from a nozzle *E* into a conical-shaped cup *C* which rotates on the end of a hollow shaft at 3000 or 4000 r.p.m. Fuel oil is conducted to the cup through a stationary pipe $\frac{1}{2}$ -inch or $\frac{3}{8}$ -inch in diameter passing through the hollow shaft of the burner. The cup rotates in a clockwise direction (as viewed from the furnace end of the burner). A strong blast of air is introduced around the cup by the fan *F*. Helical guide vanes in the air nozzle direct the air stream in a counter-clockwise whirl around and into the cone of oil spray. This whirling stream of air moving at high velocity strikes the particles of oil, breaking them up into a fine mist and propelling the mixture into the furnace. The main air supply enters the fan through the passage *D*. Secondary air to render combustion complete is drawn in through tuyeres at *G* by the injector effect of the air stream delivered by the fan, and

FIG. 78.—RAY
ROTARY OIL
BURNER



(Ray Burner Co.)

due to the draft produced by the chimney. The amount of secondary air drawn in may be controlled to suit the needs of changing loads by adjustable shutters over the tuyeres. Such adjustments may be made by hand or automatically.



(W. S. Ray Manufacturing Co.)

FIG. 79.—RAY OIL BURNER INSTALLED UNDER B. & W. BOILER

The burner cup and fan, mounted on one shaft, may be driven by a direct-connected electric motor or by a small single-stage impulse steam-turbine wheel mounted upon the burner shaft as shown in Fig. 78. A 400-horsepower burner requires about a 4 horsepower turbine to drive it. The burner units vary in size from that suitable for a small domestic heating plant to that required for a 600 horsepower boiler. For larger boilers several burners, as required, are installed. Many of these burners are

employed in marine service. The manufacturers claim among others the following advantages for the burner:

1. It will, if necessary, burn cold fuel oil;
2. It creates its own forced draft, eliminating forced draft blowers;
3. It operates on an oil pressure of from 20 to 50 lbs.;
4. The combination of low pressure and cold oil greatly reduces fire hazard;
5. The extreme range is from full capacity down to one-tenth rated capacity without adjustment;
6. It will burn the heaviest California or Mexican fuel oils efficiently;
7. It is possible to burn oil successfully containing as much as 7 per cent water.

One of the methods of installing Ray burners under Babcock and Wilcox land-type boilers is shown in Fig. 79. In this instance the burner is placed at the rear of the boiler. However, many burners are installed under the front end. But the rear-end installation gives a furnace very well adapted to the cone-shaped flame of the burner and gives a large combustion space for given furnace dimensions.¹

138. Boiler Performance. By long years of usage, the custom of expressing steam-boiler performance in terms of horsepower units has become firmly established. The term *horsepower* implies a measured force moving through a given distance in unit time. The term is correctly applied to an engine or an animal performing work. In the strict sense, a boiler does no work. But a boiler of fixed dimensions and design will generate sufficient steam to develop work at a certain rate in a steam engine of a given type and size; and work at a certain rate is horsepower. James Watt originated the horsepower unit and started the custom of designating the capacity of a boiler by the horsepower of the engine for which it is able to furnish steam. For example, a 10-horsepower boiler indicated one having the capacity to furnish steam to drive a 10-horsepower engine. There might be a

¹ For a test of a 255 horsepower Scotch marine boiler equipped with Ray burners, see Journal of the American Society of Naval Engineers, May, 1926.

reasonable excuse for designating the boiler capacity in terms of the engine horsepower it could develop if all steam engines developed a fixed quantity of power from a given weight of steam, even though such a statement is unscientific. But there is not the slightest rational basis for this custom when it is considered that the weight of steam required to generate one engine horsepower for one hour varies over wide limits in different prime movers. Furthermore, this condition will continue as time passes and the "state of the art" improves. These arguments against the use of a horsepower unit for measuring boiler performance are unanswerable on scientific grounds. But, however illogical it is, the use of the horsepower unit has become so firmly established by long use that it will be difficult to replace it by a more logical one. This question has been discussed vigorously many times and various solutions offered; but as yet no suggestion has met with general approval and adoption.

Much confusion as to the value of a boiler-horsepower unit existed in 1876. At that time, the Committee of Judges of Boiler Trials at the Centennial Exposition in Philadelphia agreed that a good engine of the type prevailing at that time required approximately 30 pounds of dry and saturated steam at a pressure of 70 pounds per square inch by gage to develop one engine-horsepower hour. It was estimated that the average temperature of feed water entering a boiler was about 100° F. To place the relation between boiler horsepower and engine horsepower upon a definite basis, the committee recommended that the evaporation of 30 pounds of feed water per hour from a temperature of 100° F. into dry and saturated steam at a pressure of 70 lbs. per sq. in. by gage be considered one boiler horsepower. In 1889, when the American Society of Mechanical Engineers formulated their first testing code, the use of the Centennial boiler unit of performance was adopted, and it is still a recommended unit of this society. This unit is in use to a considerable extent in all English-speaking countries. Every attempt to set up a more rational unit has and will continue to meet much opposition because of the inconvenience and confusion which will result when comparisons of old and new boiler performance data are to be made.

Boilers are bought and sold on the basis of the heating surface which they contain, measured in square feet. At the present time a builder's rated boiler-horsepower is considered to be 10 square feet of effective heating surface. However, such a horsepower unit is as arbitrary as the first. The quantity of heat which will flow in one hour from the hot gases of combustion through one square foot of boiler surface depends upon several factors, the more important of which are the form and extent of the metal surface; velocity and weight of hot gases; the temperature difference between the hot gases and the water in the boiler; and others. In the past few years, there has been a tendency to drop the use of the builder's rated-capacity unit, and to designate the size of all boilers by simply stating the extent of the heating surface in square feet. This is the method used altogether in European countries.

139. Boiler-Performance Computations. When, as is the case in practice, feed water is pumped into a boiler at a temperature varying between 32° F. and 212° F., or even higher, and is converted into steam at any pressure between that of the atmosphere and 3200 pounds per square inch (the critical pressure for H_2O gas), and the steam leaving the boiler may be wet, dry, or superheated, the heat content of one pound of steam is a very indefinite quantity of energy. A pound of wet steam from a small factory boiler may represent the transfer of only 1100 B.t.u. from the hot gases in the furnace through the boiler sheets into the water and steam in the boiler. Or a pound of steam leaving a high-pressure, high-superheat boiler in a modern superpower plant may carry away nearly 1400 B.t.u. But if a pound of water enters a boiler at 212° F. and is converted into steam at a pressure of 14.7 pounds per square inch absolute, 970.2 B.t.u. must be absorbed by the boiler. This performance is called *the equivalent evaporation of one pound of steam from and at 212° F.* This may be considered the fundamental unit of boiler performance.

The Centennial boiler-horsepower unit may be converted to terms of equivalent evaporation from and at 212° F. as follows: (the following numerical values are taken from *Steam Tables*, by J. H. Keenan, 1930).

The change in the heat of the liquid from 100° F. to the boiling point (316° F.) at 70 lbs. gage or 84.7 lbs. absolute equals

$$h_2 - h_1 = 286 - 68 = 218 \text{ B.t.u.}$$

The latent heat added at a pressure of 70 lbs. per sq. in. gage equals

$$r = 897.5 \text{ B.t.u.}$$

The total heat absorbed by the boiler to generate one pound of dry and saturated steam is

$$(h_2 - h_1) + r = 218 + 897.5 = 1115.5 \text{ B.t.u.}$$

The ratio

$$\frac{1115.5}{970.2} = 1.150.$$

This means that 1.150 pounds of water could have been converted into dry and saturated steam from and at 212° F. with the same quantity of heat. The value 1.150 as a ratio is called the *factor of evaporation*.

Then

$$30 \times 1.150 = 34.50.$$

Hence it is seen that the Centennial Committee definition of boiler horsepower is equal to the equivalent evaporation of 34.5 pounds of dry and saturated steam from and at 212° F. per hour. This is the unit of boiler performance in general use at the present time. In B.t.u. per hour this is equal to

$$970.2 \times 34.5 = 33\,472 \text{ B.t.u.}$$

It is sufficiently accurate to call it 33 470 B.t.u.

A new unit of boiler performance, which is slowly coming into general use, was adopted in 1923 by the Power Test Code Committee of the American Society of Mechanical Engineers. It has not yet been given a distinctive name, but is called by the cumbersome term *unit of evaporation*. A boiler develops one unit of evaporation when it absorbs 1000 B.t.u. in the generation of steam. This unit represents 1000/970.2 or 1.0307 times the heat in one pound of dry and saturated steam from and at 212° F.

The following method of computing boiler performance and efficiencies conforms with the recommendations contained in the recent Boiler Testing Code of the American Society of Mechanical

Engineers, which is universally followed by engineers in the United States.

The following symbols will be used, it being understood that the heat quantities refer to those for *one pound* of water or steam.

- a = the decimal fraction of the total steam generated which is used to atomize fuel oil,
- e_b = efficiency of the boiler in per cent,
- e_f = efficiency of the boiler, furnace, and grate for coal or efficiency of boiler, furnace, and burner for oil in per cent,
- F = factor of evaporation,
- h = heat of the liquid at boiler pressure, B.t.u., above 32° F.,
- h_f = heat of the liquid of the feed water as it enters the boiler, B.t.u. above 32° F.,
- H = total B.t.u. in 1 lb. of dry and saturated steam above 32° F. at boiler pressure,
- H_s = total B.t.u. in 1 lb. of superheated steam, above 32° F., at boiler pressure,
- H_w = total B.t.u. in 1 lb. of wet steam above 32° F. at boiler pressure
- K = pounds of dry fuel fired into the boiler furnace per hour,
- K_c = pounds of combustible actually burned per hour (for fuel oil usually $K_c = K$),
- M = pounds of feed water pumped into the boiler per hour,
- M_e = equivalent evaporation from and at 212° F., lbs. per hr.,
- r = latent heat of evaporation, B.t.u. per lb. at boiler pressure,
- t = temperature of wet or dry and saturated steam at boiler pressure, degrees F.,
- t_s = temperature of superheated steam at boiler pressure, degrees F.,
- U = calorific value of dry fuel, B.t.u. per lb. as determined by oxygen bomb calorimeter,

U_c = calorific value of combustible, B.t.u. per lb. (computed from U and the fuel analysis),

$U.E$ = unit of evaporation,

x = quality of wet steam, a decimal fraction.

For wet steam, the value of F is

$$(320) \quad F = \frac{xr + h - h_f}{970.2} = \frac{H_w - h_f}{970.2},$$

for dry and saturated steam

$$(321) \quad F = \frac{H - h_f}{970.2},$$

and for superheated steam

$$(322) \quad F = \frac{H_s - h_f}{970.2}.$$

The weight and temperature of feed water entering the boiler per hour, the steam pressure, temperature, quality, and the weight of fuel burned per hour must be determined by a boiler test. Then the following results may be computed.

$$(323) \quad M_s = MF,$$

$$(324) \quad U.E = \frac{970.2 M_s}{1000},$$

$$(325) \quad B.H.P. = \frac{M_s}{34.5}.$$

As for engines, boiler efficiency is equal to

$$(326) \quad \frac{\text{Result}}{\text{Effort}} = \frac{\text{B.t.u. absorbed by the boiler}}{\text{B.t.u. potentially in the fuel}}.$$

When the fuel is coal or oil, two boiler efficiencies are distinguished. For coal, the first is based on the weight of dry coal fired into the furnace. It is called the efficiency of the boiler furnace and grate. The second is based on the weight of combustible actually burned, and is called the efficiency of the boiler. In the sense here employed, the term *combustible* includes the weight of the wet coal minus the weights of moisture and

refuse. The refuse includes the ash as found by analysis plus any unburned coal which drops through the grate, and all soot and cinders deposited on the boiler tubes and in the passes. The boiler efficiencies are computed as follows:

$$(327) \quad e_f = \frac{970.2 \times 100 M_e}{KU} = \frac{97\,020 M_e}{KU},$$

$$(328) \quad e_b = \frac{97\,020 M_e}{K_c U_c}.$$

The value of e_b is always greater than e_f .

When oil is burned, there is no fuel loss through the grate bars, and the soot deposited on the tubes is usually negligible. But the oil has to be atomized by some agency which uses steam directly or indirectly in the form of compressed air, or mechanical sprays which are power-driven. Hence in place of efficiency of boiler furnace and grate, there is the efficiency of boiler furnace and burner. This is less than the efficiency of the boiler in proportion to the amount of atomizing steam required. Hence, the efficiencies for fuel oil are

$$(329) \quad e_f = \frac{97\,020 (M_e - aM_e)}{KU} = \frac{97\,020 M_e (1 - a)}{KU},$$

and

$$(330) \quad e_b = \frac{97\,020 M_e}{KU}.$$

Boiler performance is frequently stated in pounds of equivalent evaporation from and at 212° F. per hour per square foot of heating surface. Then, based on the rating of boiler manufacturers, the hourly evaporation of 3.45 pounds from and at 212° F. per square foot constitutes 100 per cent rated capacity. As boiler and furnace design have improved greatly in the past ten years, it is common practice to plan to have modern boilers operate normally at 150 to 250 per cent load. For short periods of peak loads, they are being driven at 500 to 800 per cent of rated load. In marine service, where weight and space is always at a premium, the tendency is toward even higher rates of driving.

140. Illustrative Test and Performance Results for a Steam Boiler.
The following data were taken during a boiler test with coal for fuel.

TABLE 41
BOILER TEST OBSERVATIONS

Type of boiler.....	Water tube
Area boiler heating surface.....	6 000 sq. ft.
Area grate surface.....	132 sq. ft.
Weight of wet coal fired per hour.....	3 810 lbs.
Weight of refuse per hour.....	375 lbs.
Weight of feed water per hour (M).....	32 800 lbs.
Steam pressure in boiler, lbs. per sq. in. gage.....	200
Steam temperature.....	487.9° F.
Feed-water temperature.....	57° F.
Stack-gas temperature.....	500° F.
Boiler-room temperature.....	80° F.
Barometer reading.....	29.90 in. Hg.

TABLE 42
PROXIMATE ANALYSIS OF COAL USED IN BOILER TEST
Classification of Coal; Semibituminous

	Wet Coal 100%	Dry Coal 100%	Com- bustible 100%
Moisture.....	4.0		
Volatile matter.....	21.8	22.7	24.4
Fixed carbon.....	67.6	70.5	75.6
Ash.....	6.6	6.8	
Total.....	100.0	100.0	100.0
Calorific value B.t.u. per lb.....	14 050	14 635	15 720

TABLE 43
ULTIMATE ANALYSIS
Dry Coal, 100 per Cent
(Per Cents by Weight)

Carbon.....	79.2
Hydrogen.....	5.1
Sulphur.....	2.1
Nitrogen.....	1.3
Oxygen.....	5.5
Ash.....	6.8
Total.....	100.0

TABLE 44
FLUE GAS ANALYSIS
(Per Cents by Volume)

Carbon dioxide.....	13.5
Oxygen.....	5.2
Carbon monoxide.....	0.2
Nitrogen (by difference).....	81.1
Total.....	100.0

COMPUTED RESULTS OF BOILER TEST

Absolute pressure of the steam in the boiler equals

$$p_1 = 200 + (29.90 \times .491) = 214.7 \text{ lbs. per sq. in.}$$

Referring to a table of properties of superheated steam, it is observed that at a pressure of 214.7 lbs. per sq. in. absolute and for a temperature of 487.9° F., the steam is superheated 100° F. Therefore the value of the factor of evaporation is

$$F = \frac{H_s - h_f}{970.2} = \frac{1258.9 - 25.1}{970.2} = 1.2717$$

The equivalent evaporation from and at 212° F. per hour is

$$M_e = 32\,800 \times 1.2717 = 41\,718 \text{ lbs.}$$

The units of evaporation are

$$U.E. = \frac{970.2 \times 41\,718}{1000} = 40\,475.$$

The pounds of dry coal fired per hour are

$$K = 3810 \times 0.96 = 3658 \text{ lbs.}$$

The heat in 1 lb. of dry coal is

$$U = \frac{14\,050}{1.00 - 0.04} = \frac{14\,050}{0.96} = 14\,635 \text{ B.t.u.}$$

By definition, combustible equals the sum of the volatile matter and fixed carbon as determined by proximate analysis. Hence the combustible in 1 lb. of wet coal is found to be (from Table 42)

$$21.8\% + 67.6\% = 89.4\% \text{ or } 0.894 \text{ lbs.}$$

The heat per lb. of combustible is

$$U_c = \frac{14\,050}{0.894} = 15\,720 \text{ B.t.u.}$$

The pounds of combustible burned per hour equals the weight of wet coal fired minus the refuse and the moisture. Therefore

$$K_c = K - \text{refuse} = 3658 - 375 = 3283 \text{ lbs.}$$

By (327), the efficiency of the boiler, furnace, and grate is

$$e_f = \frac{97\,020 \times 41\,718}{14\,635 \times 3658} = 75.6\%,$$

and by (328), the efficiency of the boiler is

$$e_b = \frac{97\,020 \times 41\,718}{15\,720 \times 3283} = 78.4\%.$$

The builders' rated capacity is

$$\frac{6000}{10} = 600 \text{ B.hp.}$$

The developed boiler horsepower is

$$\text{B.hp.} = \frac{41\,718}{34.5} = 1209.$$

The rate of driving or the percentage of builders' rated capacity developed is

$$\frac{1209 \times 100}{600} = 201.5\%.$$

The equivalent evaporation from and at 212° F. per hour per square foot of boiler heating surface is

$$\frac{41\,718}{6000} = 6.95 \text{ lbs.}$$

The equivalent evaporation from and at 212° F. per lb. of dry coal fired is

$$M_k = \frac{41\,718}{3658} = 11.4 \text{ lbs.}$$

The weight of dry coal fired per square foot of grate surface per hour is

$$\frac{3658}{132} = 27.7$$

141. Boiler Heat Balance.¹ It is interesting and profitable to make a detailed analysis of the disposition of the heat liberated when a pound of coal is burned in a boiler furnace. From 50 per cent to 90 per cent of the heat is absorbed by the boiler and auxiliary heat-absorbing equipment. What becomes of the remainder of the heat? The answers to that question are very important aids in the effort to improve boiler performance.

In the Boiler Test Code of the American Society of Mechanical Engineers, eight items are differentiated in the simplest form of the boiler heat balance for a boiler without an economizer. The quantities

¹ For graphical method of determining boiler heat balance, see Transactions, A.S.M.E., vol. 50, 1928, FSP-paper, No. 80.

involved are represented by symbols, some of which have been explained on page 207. The others are explained below.

B = B.t.u. absorbed by the boiler per lb. of dry fuel fired,

C_b = lbs. of carbon burned per lb. of dry fuel fired,

C_r = lbs. of unburned carbon in the refuse per lb. of dry fuel fired,

h_r = heat of the liquid per lb. of water at room temperature, B.t.u.,

H_a = total heat in 1 lb. of dry and saturated steam at atmospheric pressure (about 1150 B.t.u.),

L = with appropriate subscript, the various heat losses,

M_g = lbs. of dry chimney gases per lb. of dry fuel burned,

M_k = equivalent evaporation from and at 212° F. per lb. of dry coal fired,

M_w = lbs. of moisture entering the furnace in the coal with each lb. of dry coal fired,

t_c = temperature of the coal as it enters the furnace, degrees F.,

t_g = temperature of the gases as they leave the boiler, degrees F.,

t_r = temperature of the air just before entering the furnace degrees F. (This temperature is generally considered to be the same as the room temperature, unless the air is preheated).

Using the data and results from the illustrative boiler test of § 140, the method of computing a boiler heat balance is given below. *The heat balance is generally based upon the calorific value of 1 pound of dry coal, as determined by calorimeter, considered as 100 per cent.*

(1). The heat absorbed by water and steam in the boiler is

$$(331) \quad B = 970.2 M_k = 970.2 \times 11.4 = 11\,060 \text{ B.t.u.}$$

In percentage, this is

$$\frac{11\,060 \times 100}{14\,635} = 75.60\%.$$

Compare this value with the value of e_f on page 271.

(2). The loss due to evaporation and superheating the moisture accompanying 1 lb. of dry fuel into the furnace is

$$(332) \quad L_1 = M_w[H_a - h_r + 0.47(t_g - 212)].$$

The A.S.M.E. Test Code Committee proposes the approximate formula stated in (333) which is somewhat simpler to solve and it gives

practically the same result as (332):

$$(333) \quad L_1 = M_w(1089 + 0.46 t_g - t_c).$$

From Table 42 it is seen that

$$M_w = \frac{0.04}{0.96} = 0.04166.$$

Hence

$$L_1 = 0.04166 [1150.2 - 48 + 0.47 (500 - 212)] = 51 \text{ B.t.u.}$$

In percentage,

$$L_1 = \frac{51 \times 100}{14\,635} = 0.35\%.$$

(3). The loss due to the heat carried away in the uncondensed water of combustion is

$$(334) \quad L_2 = 9 M_h [H_a - h_r + 0.47 (t_g - 212)].$$

The bracket in (334) may be replaced by the parenthesis of (333), as may be done in finding L_1 :

$$L_2 = 9 \times .051 [1150 - 48 + 0.47 (500 - 212)] = 568 \text{ B.t.u.}$$

In percentage,

$$L_2 = \frac{568 \times 100}{14\,635} = 3.88\%.$$

(4). The loss due to the heat carried away in the dry chimney gases is

$$(335) \quad L_3 = 0.24 M_g (t_g - t_r)$$

$$(336) \quad M_g = C_b \left[\frac{11 \text{ CO}_2 + 8 \text{ O}_2 + 7 (\text{CO} + \text{N})}{3 (\text{CO}_2 + \text{CO})} \right].$$

See page 207 above for explanation of the symbols within the brackets in (336).

From the test data the refuse equals

$$\frac{375 \times 100}{3658} = 10.25\% \text{ of the dry coal fired.}$$

By analysis the ash is 6.8% of the dry coal fired. Therefore the percentage of unburned carbon is

$$100 C_r = 10.25 - 6.8 = 3.45\%,$$

and

$$C_r = 0.0345 \text{ lbs.}$$

Then from the ultimate analysis, the carbon actually burned is

$$C_b = 0.792 - 0.0345 = 0.7575 \text{ lbs.}$$

Therefore (336) becomes

$$M_g = 0.7575 \left[\frac{11 \times 13.5 + 8 \times 5.2 + 7 (0.2 + 81.1)}{3 (13.5 + 0.2)} \right] = 13.99 \text{ lbs.},$$

and (335) becomes

$$L_3 = 0.24 \times 13.99 (500 - 80) = 1410 \text{ B.t.u.}$$

In percentage,

$$L_3 = \frac{1410 \times 100}{14\,635} = 9.64\%.$$

(5). The loss due to incomplete combustion of carbon with the formation of carbon monoxide is

$$(337) \quad L_4 = C_b \left(\frac{10\,250 \text{ CO}}{\text{CO}_2 + \text{CO}} \right).$$

Substituting the appropriate test data the result is

$$L_4 = 0.7575 \left(\frac{10\,250 \times 0.2}{13.7} \right) = 113 \text{ B.t.u.}$$

In percentage,

$$L_4 = \frac{113 \times 100}{14\,635} = 0.77\%.$$

(6). The loss due to unconsumed carbon in the refuse equals L_5 . For simplicity it is assumed that all the unburned combustible which falls into the ash pit or collects on the boiler tubes and in the various passages is pure carbon. This assumption is not strictly accurate, but it is sufficiently accurate for most purposes. If an accurate result is desired, it is necessary to thoroughly mix all the refuse, sample it, and from the sample determine the calorific value of it as is done for the fuel. The calorific value of a pound of pure carbon is taken as 14 600 B.t.u. Then

$$(338) \quad L_5 = 14\,600 C_r.$$

In computing the value of M_g in (336) above, the value of C_r was found to be 0.0345 lbs. Therefore

$$L_5 = 14\,600 \times 0.0345 = 504 \text{ B.t.u.}$$

In percentage,

$$L_5 = \frac{504 \times 100}{14\,635} = 3.44\%.$$

(7). The loss due to superheating the water vapor carried into the furnace with the air supply, L_6 , is usually about 0.5 per cent of the

heating value of the dry fuel. This item may be neglected without serious error. Before it can be computed, a chapter on saturated, partially saturated, and dry air would have to be studied. Such a chapter seems out of place in an elementary text on Heat Engines.¹ Therefore L_6 is estimated to be $0.3\% = 44$ B.t.u. This is not greatly different from the true value correctly determined.

(8). The loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted for, equals L_7 . It is found by difference as follows:

$$(339) \quad L_7 = U - (B + L_1 + L_2 + L_3 + L_4 + L_5 + L_6).$$

Therefore for the illustrated problem L_7 is

$$\begin{aligned} L_7 &= 14,635 - (11,060 + 51 + 568 + 1410 + 113 + 504 + 44) \\ &= 885 \text{ B.t.u.} \end{aligned}$$

In percentage

$$L_7 = \frac{885 \times 100}{14\,635} = 6.02\%.$$

The items of the boiler heat balance are generally arranged as shown in Table 45.

TABLE 45
BOILER HEAT BALANCE—SHORT FORM *

ITEM No.	ITEM	DRY COAL = 100%	
		B.t.u.	Per Cent
1	Heat absorbed by the boiler.....	11 060	75.60
2	Loss due evaporation of moisture in fuel.....	51	0.35
3	Loss due to heat carried by steam formed by burning of hydrogen.....	568	3.88
4	Loss due to heat carried away in dry flue gases.....	1 410	9.64
5	Loss due to incomplete combustion of carbon.....	113	0.77
6	Loss due to unconsumed combustion in the refuse....	504	3.44
7	Loss due to heating moisture in the air.....	44	0.30
8	Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for.....	885	6.02
	Total calorific value of 1 lb. of dry fuel.....	14 635	100.00

* See *Test Code for Stationary Steam Boilers* by the American Society of Mechanical Engineers, 1923, for more complete forms for boilers equipped with economizers, superheaters, and air preheaters.

¹ See G. F. Gebhardt, *Steam Power Plant Engineering*, for a brief chapter on this subject; or Transactions A.S.M.E., vol. 33, 1911, p. 1005.

In general, the values of the important boiler losses fall within the limits stated in Table 46.

TABLE 46
VALUES OF THE VARIOUS BOILER LOSSES

Item	Per Cent Loss
Loss in dry chimney gases.....	5 to 25
Loss by radiation and unaccounted for.....	2 to 15
Loss due to water from combustion of hydrogen...	2 to 6
Loss due to incomplete combustion of carbon.....	0 to 3

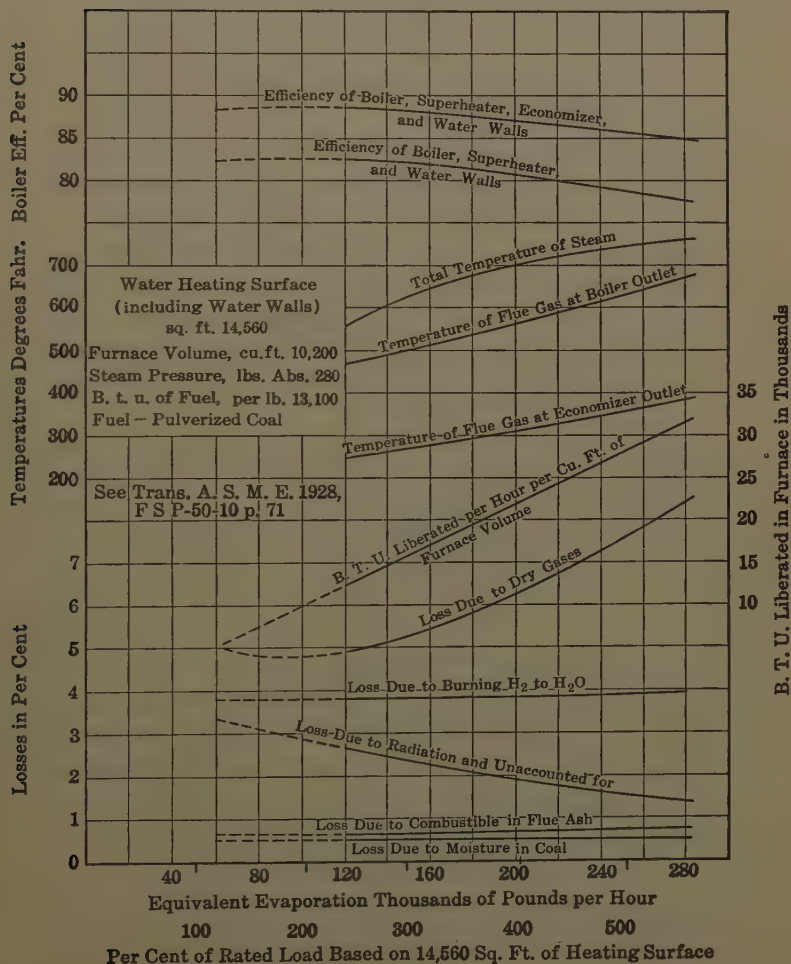


FIG. 80.—BOILER-PERFORMANCE CURVES

Performance results of notable boiler installations will be found in Fig. 80 and the references listed below.¹

142. Boiler Efficiencies. The item of boiler performance which is considered most important is boiler efficiency. The range of values of efficiency varies from a low value of 30 or 40 per cent, in small industrial plants using a single boiler of from 10 to 75 horsepower, to 90 per cent in large modern central stations employing boilers of the highest type of design, equipped with highly efficient automatic control and auxiliary heat-absorbing devices, and operated by a highly trained corps of engineers and firemen. This upper value is within 4 or 5 per cent of the theoretical maximum which is attainable.

The efficiency of a boiler depends upon many factors, the more important of which are large temperature difference between the gases and the water in the boiler, the size of the unit, the rate of driving, the arrangement of the boiler surfaces, the dimensions and form of the furnace, the velocity of the hot gases over the heating surfaces, the velocity and thoroughness of the water circulation in the boiler, the kind and quality of fuel, the distribution and control of the air supply in such a way as to develop as nearly as possible the theoretical percentage of CO_2 without formation of CO , air-tight furnace walls, and the care and skill used in boiler operation. From this extensive list of variables it is seen that the problem of so combining all the factors as to obtain the highest possible boiler efficiencies in any given installation is not a simple one.

143. Boiler Pressures. In the days of Watt, boiler pressures in use were just about that of the atmosphere or slightly greater. From that time forward, there has been a continued trend of rising pressures. Since 1920, there has been a very marked rise in pressures employed in the modern, central power stations which have been put into operation.

At the present time, the prevailing pressures in small power plants are from 100 to 150 pounds per square inch gage. In

¹ Transactions, A.S.M.E., 1923, p. 567; 1924, p. 595; 1926, pp. 119, 1319, and 1380; vol. 50, 1928, FSP. 50-10 (71), and FSP. 50-34 (83); Proceedings N.E.L.A., 1927, pp. 912 and 920.

small and moderate-sized central stations built ten to fifteen years ago, and in locomotive and marine service, pressures vary from 200 to 300 pounds. In modern central power stations built since 1920, pressures of 400, 600, and 1400 pounds per square inch are employed with sufficient superheating to bring the temperature of the steam to a maximum of 750° F. or 800° F. In England the Benson boiler, which generates steam at a pressure of 3200 pounds per square inch (the critical pressure of steam) is being tried out on an experimental basis.¹ At the critical pressure the latent heat of evaporation is zero and the specific volumes of the liquid and the vapor are equal. The chief advantage of generating steam at the critical pressure is that ebullition and priming are eliminated. In the Benson system, the steam is throttled down to about 1500 pounds pressure per square inch with some additional superheating before it enters the engine or turbine.

Present trends in the generation of steam are strongly toward higher pressures, with the critical pressure as the probable upper limit. But indications seem to forecast that pressures much above 1500 pounds per square inch will not be employed for the generation of power in this country for some years.

144. Boiler Efficiency and CO₂ Content of Flue Gases. Low percentages of CO₂ in the flue gases indicate large excess weights of air passing through the furnace, or serious leakage of air through the furnace setting. If the furnace is small and unsuited for the type of fuel burned, it is impossible to obtain high percentages of CO₂ without generating considerable CO. With amply and correctly designed furnaces equipped with stokers or burning pulverized coal, the standard of good performance is from 15 to 16 per cent of CO₂ by volume in the dry flue gases. This result is attained with practically no carbon monoxide in the flue gas.

The theoretical fuel saving possible when the CO₂ content of flue gas increases may be readily computed. But a new calculation is necessary for every change in fuel analysis and for different

¹ See Power, pp. 796 and 842, May 22 and 29, 1923; pp. 454 and 457, Mar. 18, 1924.

Power Plant Engineering, p. 740, July 1, 1927.

temperatures of the flue gas if accurate results are desired. However, an idea of the magnitude of the savings possible may be obtained by making certain assumptions. Table 47 has been computed for pure carbon as fuel, a constant stack temperature of 500° F., and zero CO for all percentages of CO₂. It is also assumed that 16 per cent of CO₂ cannot be exceeded in practice without production of appreciable amounts of CO. Upon that basis, the boiler efficiency would theoretically be zero when the CO₂ content falls to 1.5 per cent. The values of efficiency in the table are nearly correct for an anthracite coal when burned in a modern and skillfully operated boiler plant. The order of magnitude of the tabular values are more or less approximately correct for other fuels.

TABLE 47

THEORETICAL FUEL SAVINGS FOR INCREASES IN CO₂ IN FLUE GASES¹

Increase in CO ₂ per Cent by Volume in Dry Gas	Theoretical Fuel Savings per Cent	Total Possible Savings in % if CO ₂ Were Raised to 16 per Cent	Percentage of Excess Air at Lower CO ₂ Value
2 to 3	28.7	74.1	935
3 to 4	14.0	45.4	590
4 to 5	8.5	31.4	417
5 to 6	5.5	22.9	314
6 to 7	4.0	17.4	245
7 to 8	3.0	13.4	195
8 to 9	2.3	10.4	159
9 to 10	1.9	8.1	130
10 to 11	1.6	6.2	107
11 to 12	1.3	4.6	88
12 to 13	1.1	3.3	72
13 to 14	0.9	2.2	59
14 to 15	0.8	1.3	48
15 to 16	0.5	0.5	38
16 to 20	0.0	0.0	30

A study of the above table shows that the relations between the fuel saving possible for one per cent increase in CO₂ varies

¹ See Joseph W. Hays, *How to Build Up Furnace Efficiency*, p. 496. Seventeenth edition, 1924.

approximately as the following exponential equation:

$$(340) \quad S(\text{CO}_2)^{1.83} = 104.5.$$

S = per cent of fuel saved by increasing CO_2 one per cent by volume in dry flue gas.

CO_2 = Lower per cent by volume of CO_2 in the dry flue gas.

Equation (340) indicates the importance of obtaining at all times as nearly as possible 16 per cent of CO_2 in the flue gas.

It is necessary to have some excess air with all fuels, but for gas, oil, and pulverized coal, the excess can be smaller than for coals, stoker-fired, without undue production of carbon monoxide. Table 48 shows the lower values of excess air which are possible with maintenance of satisfactory combustion.

TABLE 48
MAXIMUM CO_2 AND MINIMUM AIR SUPPLY

Fuel	Max. CO_2 per Cent	Max. CO_2 without CO , per Cent	Min. Excess Air Practical, per Cent
Pure carbon	20.8	16.	30.
Bituminous coal, stoker fired. . .	18.6	15.	20.
Pulverized coal.	18.6	16.	15.
Fuel oil	15.6	14.	10.
Natural gas.	11.8	11.	5.

145. Boiler Efficiency and Furnace Temperature. The rate of heat transfer, by combined radiation and convection obtaining in a boiler furnace, varies approximately as the square of the temperature difference between the hot gases and the water in the boiler. Heat transfer is also proportional to the velocity of the hot gases flowing over, or through, the boiler tubes, to the conductivity of the metal, and inversely proportional to the metal thickness. It is thus seen that it is very important that the temperature of the hot gases shall be as high as possible in order to produce high rates and high efficiency of heat transfer. In order to withstand the high temperatures which can be obtained in large furnaces burning pulverized coal and the better type of stokers or with fuel oil, it is necessary to provide some means of

cooling the furnace walls, as by water screen placed inside and adjacent to the walls, or by preheating air-ducts built into them. The more usual method is to employ water screens, as already described. Maximum furnace temperatures of 2800° F. in the large boiler furnaces are frequently obtained in practice. Furnace temperatures of smallish and average industrial plants vary usually from 2000° F. to 2500° F.

146. Boiler Efficiency and Velocity of Water Circulation.

Performance results of modern locomotive boilers and marine express boilers used in torpedo-boat destroyers clearly indicate the importance of a rapid and thorough circulation of water if high boiler capacities with high efficiency are to be obtained.

Steam or gas are poor conductors of heat compared with metals or water. As steam is generated in a boiler it gathers in small bubbles which adhere to the boiler surface. Unless the bubbles are removed as fast as generated, a nearly continuous film of steam gathers upon the boiler plates. This film of steam materially reduces the rate of heat transmission into the water. The resistance of a film of steam to the passage of heat is 50 times greater, or even more, than for an equal thickness of metal. The only way to prevent the serious retardation of heat flow by the steam bubbles is to remove them as rapidly as they are formed. The only effective method of doing this is to insure constant and rapid circulation of the water. The velocity of the water must be high enough to detach every steam bubble as fast as it forms, overcoming the force of adhesion holding the bubble to the plate, and sweeping them along into the steam space where water and steam separation takes place. This will insure at all times that the boiler plates are completely covered with water. Tests have indicated that under the conditions prevailing in a modern boiler and furnace, boiler plates completely covered with water never reach a temperature more than 50° to 100° F. higher than the water. The velocity of water circulation over the various active surfaces of a boiler varies widely depending upon the form and location of the surfaces. Velocities varying from less than 1 foot per second to 8 or 10 feet per second have been measured. The

lower values are not high enough to give a rate of heat transfer at reasonable efficiency necessary to produce 100 per cent of builders' rated capacity. The velocity should be not less than 2 to 4 feet per second at the rated capacity of the boiler, and for ratings of 300 or 400 per cent frequently obtained in practice the velocity should be 6 to 8 feet per second.

PROBLEMS

1. Compute the factor of evaporation for the following conditions: boiler pressure 150 lbs. per sq. in. by gage, steam temperature 465.9° F., feed water temperature 180° F., barometer reading 29.65 in. of mercury.

2. A boiler generates 3000 lbs. of steam per hour under the conditions given in Problem 1. What is the total equivalent evaporation from and at 212° F. per hour?

3. The following data was taken during a boiler test: Mean boiler pressure, 175 lbs. per sq. in. gage; mean temperature of the steam, 477.5° F.; temperature of the feed water, 190° F.; weight of water fed to the boiler, 46 500 lbs. per hr.; weight of fuel oil burned, 3680 lbs. per hr.; calorific value of fuel oil, 18 600 B.t.u. per lb. of dry oil; steam used to atomize the oil, 900 lbs. per hr.; barometer reading, 29.45 in. of mercury. Compute: (a) factor of evaporation, (b) boiler horsepower developed, (c) efficiency of boiler, furnace, and burner, (d) efficiency of boiler.

4. If the efficiency of boiler, furnace, and grate is 0.75, and wet coal whose calorific value is 13 600 B.t.u. containing 4.5 % of moisture costs \$3.50 per ton of 2000 lbs.; (a) what is the fuel cost for 1000 lbs. of steam at a pressure of 150 lbs. per sq. in. abs. and 150 degrees of superheat with a feed water temperature of 150° F.? What would be the cost if all other conditions were as above, but the feed water temperature were 60° F.? (b) What would be the cost if the feed water temperature were 250° F.? (c) What would be the cost if all the conditions were as in (a) except that the boiler efficiency equaled 0.85?

5. A steam turbine requires 15 lbs. of steam per kilowatt-hour delivered at the switchboard. The steam is supplied at a pressure of 200 lbs. per sq. in. abs. and a temperature of 501.9° F. The temperature of the feed water is 180° F. The boiler efficiency is 78 per cent. How many pounds of wet coal are required per kilowatt-hour when the calorific value of the wet coal is 13 500 B.t.u. per lb.?

6. The following data was taken during a 24-hour test of a Babcock and Wilcox cross-drum boiler containing 15 326 sq. ft. of heating surface:¹ boiler pressure 321 lbs. per sq. in. abs., temperature of the steam leaving the superheater 571° F., temperature of the feed water 94° F., temperature of the hot gases leaving the boiler 555° F., temperature of air entering the furnace 59° F., weight of feed water 89 158 lbs. per hour, weight of wet coal fired 10 183 per hour, the coal being in the pulverized form. Flue-gas analysis in per cents by volume is CO₂ 12.9%; O₂ 6.1%; CO 0.06%. There was no unburned carbon in the refuse.

¹ See Transactions A.S.M.E., vol. 46, 1924, p. 608.

ULTIMATE ANALYSIS OF COAL, PER CENTS BY WEIGHT		PROXIMATE ANALYSIS, PER CENTS BY WEIGHT	
Moisture.....	1.51	Moisture.....	1.5
Carbon.....	73.44	Volatile matter.....	35.0
Hydrogen.....	4.76	Fixed carbon.....	51.6
Oxygen.....	6.28	Ash.....	11.9
Nitrogen.....	1.45		
Sulphur.....	0.71	Total.....	100.0
Ash.....	11.85	B.t.u. per lb. of wet coal =	13 246.
Total.....	100.00		

Compute (a) boiler horsepower developed, (b) percentage of builders' rated capacity developed, (c) the efficiency of the boiler.

7. Compute a heat balance for the combined boiler and superheater from the data given in Problem 6.

8. A water-tube boiler with superheater is desired to deliver 100 000 lbs. of steam per hr. under the following conditions: steam pressure 350 lbs. per sq. in. gage, steam temperature 550° F., feed-water temperature 200° F. The boiler is to be operated at 200 per cent of rated capacity. Determine (a) the factor of evaporation; (b) the required area of active heating surface exclusive of superheater.

9. Coal whose analysis is given under Problem 6 was burned under a boiler with 40 per cent excess air, no CO being formed. The unburned carbon in the refuse was 1.5 per cent of the coal fired. The temperature of the flue gases leaving the boiler was 550° F. Compute theoretical flue-gas analysis and heat balance.

10. Using the boiler efficiency and coal analysis from Problem 6, determine the lbs. of coal per hour required by the boiler of Problem 8.

CHAPTER XIV

STEAM ENGINES

147. Introduction. The steam engine was the first power machine employed in transforming heat energy into useful work on a commercial scale. The first engine employing a reciprocating piston in a cylinder was invented by Dionysius Papin, a French scientist, in 1690.¹

The generally prevailing notion that James Watt invented the steam engine is erroneous. Edward Somerset, second Marquis of Worcester, Papin, and Thomas Savery all made important and basic contributions to its development. Thomas Newcomen was probably the first to combine in a coördinated workable power plant the steam boiler, reciprocating piston, and cylinder. He was the first to connect the piston to an oscillating beam, thus producing the general design of the steam engine which remained the standard form for approximately 150 years. Fig. 81 shows one of the early Newcomen engines with the cylinder placed directly over the boiler, and with the piston and pump rods attached to opposite ends of the walking beam. Newcomen's first commercial engine probably began its work in 1712, pumping water from a coal mine near Birmingham.² The operation of the engine has been described as follows:³

"The steam was generated in a separate boiler, as in Savery's engine, from which it was conveyed into a vertical cylinder underneath a piston fitting it closely, but moveable upwards and downwards through its whole length. The piston was fixed to a

¹ See the following:

R. S. Stuart, *History of the Steam Engine*.

John Farey, *Treatise on The Steam Engine*.

Thomas Tredgold, *The Steam Engine*.

Elijah Galloway, *A History of the Steam Engine*.

Samuel Smiles, *Lives of the Engineers*.

W. J. M. Rankine, *The Steam Engine*.

R. H. Thurston, *History of the Steam Engine*.

J. A. Ewing, *The Steam Engine and Other Heat Engines*.

² See Transactions of the Newcomen Society.

³ Samuel Smiles, *Lives of the Engineers*, Vol. 4, p. 63.

rod, which was attached by a joint or a chain to the end of a lever vibrating upon an axis, the other end being attached to a rod working a pump. When the piston in the cylinder was raised, steam was let into the vacated space through a tube fitted into the top of the boiler, and mounted with a stopcock. The pump-rod, at the further end of the lever being thus depressed,

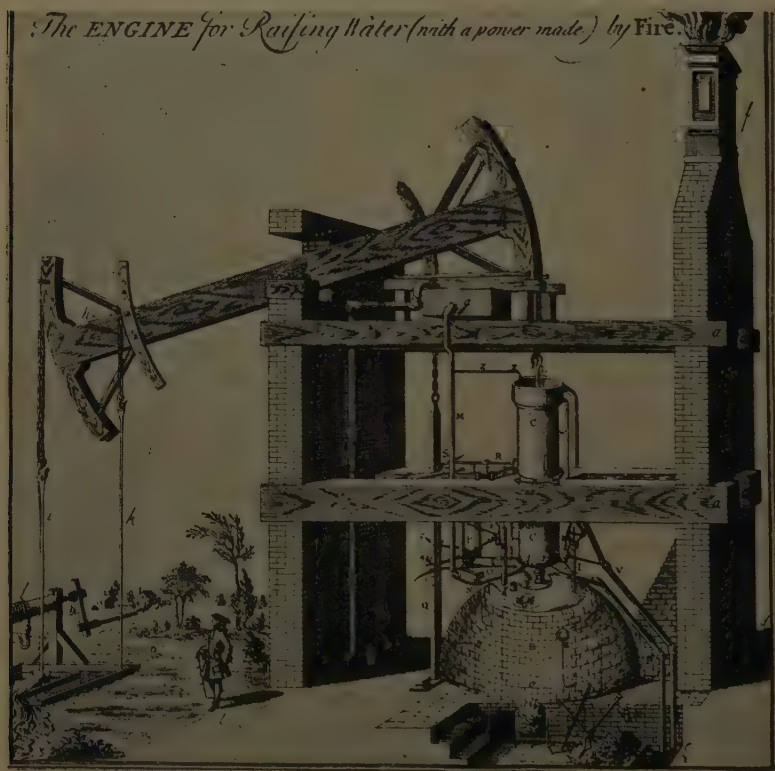


FIG. 81.—NEWCOMEN ENGINE DRAWN BY BEIGHTON IN 1717

cold water was applied to the sides of the cylinder, on which the steam within was condensed, a vacuum was produced, and the external air, pressing upon the top of the piston, forced it down into the empty cylinder. The pump-rod was thereby raised; and the operation of depressing and raising it being repeated, a power was thus produced which kept the pump continuously at

Lettered Parts:

- a*, boiler and setting
- b*, steam main
- c*, engine cylinder
- d*, anchor bolts
- e*, cylinder head and stuffing box
- fg*, walking beam
- j*, main pump rod
- k*, piston rod
- m*, condensing cistern
- n*, air pump
- o*, condenser
- p*, cold water pump
- r, s, t*, valves
- u*, boiler feed pump
- x*, feed-water valve
- y*, float control on feed-water valve

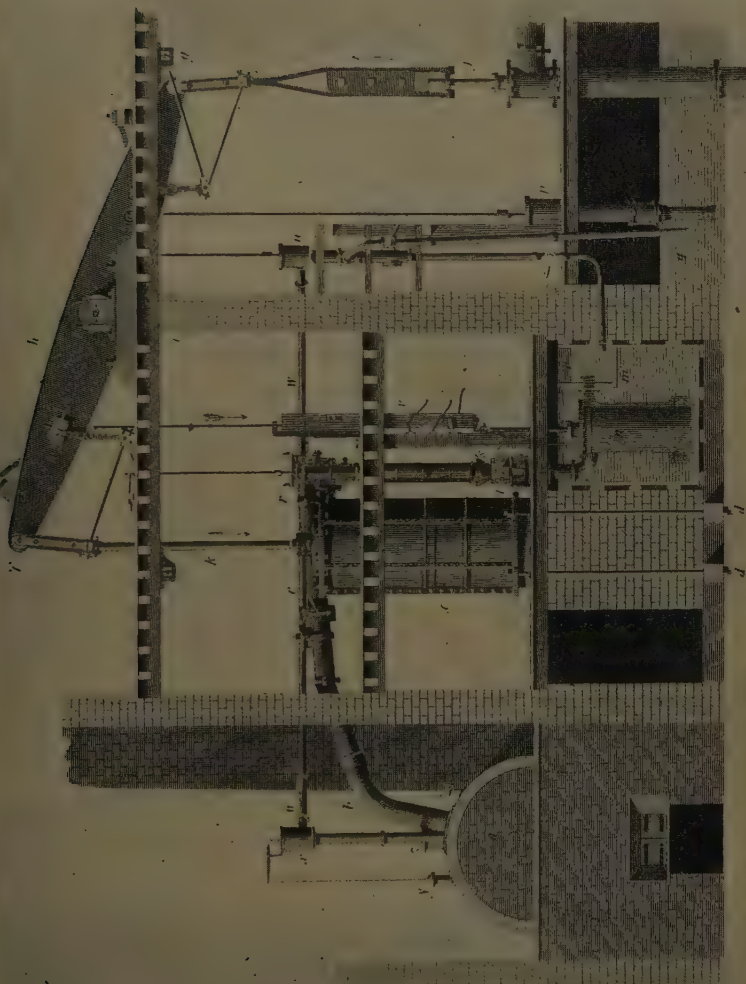


FIG. 82.—BOULTON AND WATT SINGLE-ACTING STEAM ENGINE (Model of about 1790)

work. Such, in a few words, was the construction and action of Newcomen's first engine."

To keep the early Newcomen engines at work, the steam and cold-water cocks were hand-operated. This was an easy job usually assigned to a boy. It was, however, a very confining and monotonous duty requiring constant attention. Tradition has it that one bright and restless boy, Humphrey Potter, who was assigned to this work for one of the engines, noticed the alternating motions of the beam. He conceived the idea of an arrangement of strings operated from the beam which accomplished the necessary valve movements. He called the contrivance a *scogan*, a term in those times meaning to shirk one's work. Although this improvement came from the desire and inclination of a boy to play, it made the engine automatic and improved its working power, the number of effective strokes being increased from 6 or 8 to about 15 per minute. This invention constituted the beginning of a long line and great variety of valve gears. Potter grew up to be a skilled mechanic employed in erecting many Newcomen engines. Newcomen died in 1729, and was buried in Bunham Fields, London, where lie the remains of Defoe and Bunyan.

148. Classification of Steam Engines. Engines may be classified in several ways. Certain of them are based upon structural features and others refer to the manner of operation:

1. As to the type of service:

- a) Stationary engines
- b) Locomotive engines
- c) Marine engines
- d) Heavy-duty mill engines
- e) Blowing engines "
- f) Pumping engines
- g) Hoisting engines
- h) Automotive engines.

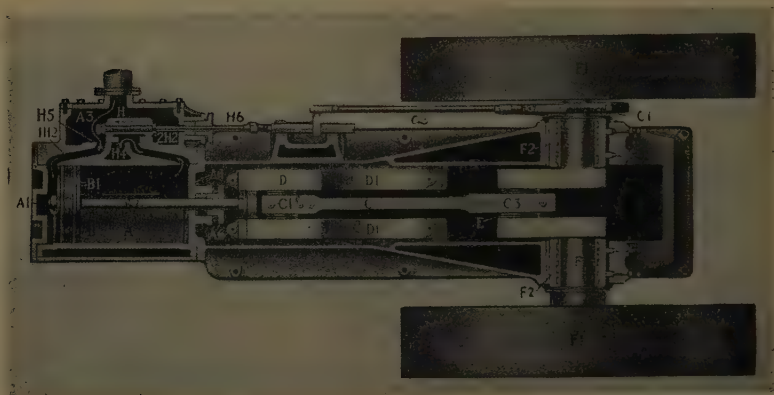
2. As to the position of the cylinder axis:

- a) Vertical
- b) Horizontal

- c) Angle
 - d) Combination of (a) and (b).
3. As to the number of cylinders in which the progressive expansion of the steam occurs:
 - a) Simple engine
 - b) Compound engine
 - c) Triple-expansion and multiexpansion engines.
 4. As to the type of valves and valve gears:
 - a) Slide valve
 - b) Piston valve
 - c) Poppet valve, or lift valve
 - d) Corliss, or rocking valve.
 5. As to the number of power impulses per revolution:
 - a) Single-acting
 - b) Double-acting.
 6. As to the path of the steam through the cylinder:
 - a) Return-flow, or contraflow
 - b) Straight-line flow, or unaflow.
 7. As to rotative speed of the engine shaft:
 - a) Low-speed, below 150 revolutions per minute
 - b) Medium-speed, approximately 150 to 250 r.p.m.
 - c) High-speed, above 250 r.p.m.
 8. As to method of governing the speed:
 - a) Automatic throttling governor
 - b) Automatic cut-off governor
 - c) Hand-throttle control
 - d) Combination of (a) or (b) with (c).
 9. As to the exhaust pressure:
 - a) Non-condensing, exhausting to the atmosphere
 - b) Condensing, exhausting into a condenser where the pressure is considerably below atmospheric.

149. Structural Features of Steam Engines. A sectionalized view of a small modern horizontal steam engine is shown in Fig. 83, which illustrates the essential features of all engines. They are: a cylinder *A*, piston *B*, piston rod *B7*, connecting rod *C*,

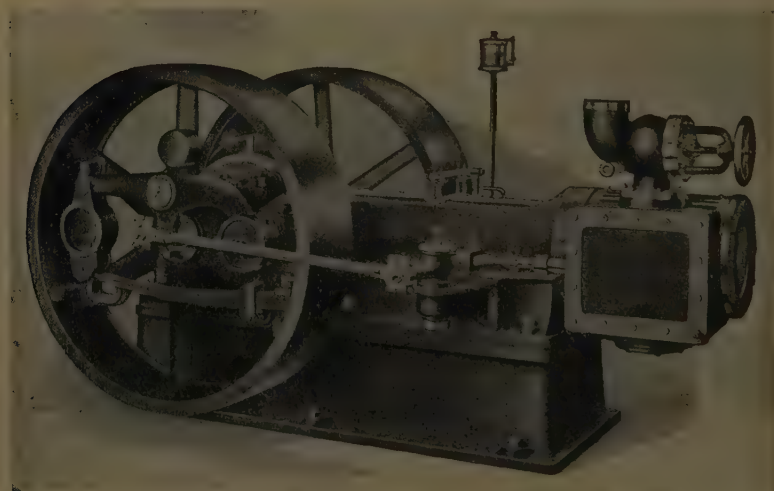
crankshaft *F*, crosshead *D*, crosshead guides *D1*, valve chest *A3*, slide valve *H*, eccentric *G1*, flywheels *F1*, and eccentric rods



(Vacuum Oil Co.)

FIG. 83.—SIMPLE SLIDE-VALVE STEAM ENGINE

G3 and *H6*. These various parts are attached to a base which holds them rigidly in their respective positions. The piston,



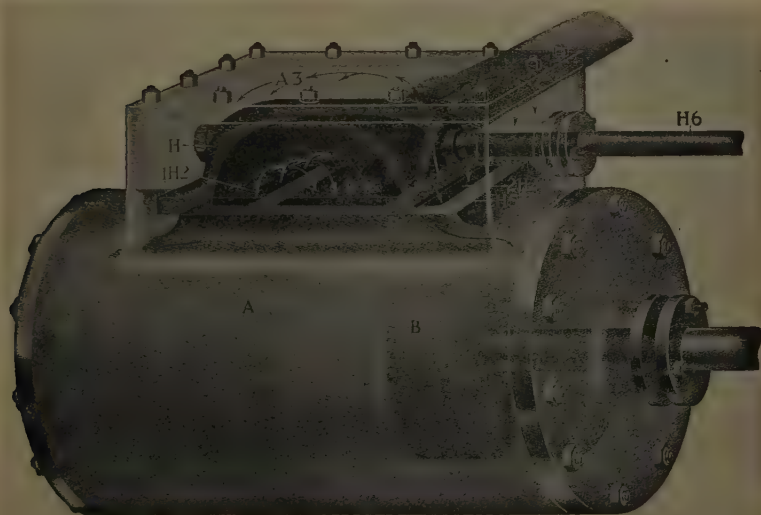
(Vacuum Oil Co.)

FIG. 84.—HIGH-SPEED ENGINE WITH FLYWHEEL GOVERNOR

piston rod, crosshead, connecting rod, crankshaft, and flywheels, together with the valve, valve rods, and eccentric constitute the

moving system of a steam engine. The cylinder, base, and cross-head guides are fixed and stationary. Fig. 84 is a photographic view of the steam engine corresponding to the sectionalized view of Fig. 83. These figures represent a small high-speed slide-valve engine built in sizes which generally range from about 10 horsepower up to 200 or possibly 300 horsepower.

The end of the cylinder adjacent to the base is called the *crank end* and the opposite end is called the *head end*. The reciprocating piston *B* is equipped with rings *B1* which fit closely in



(Vacuum Oil Co.)

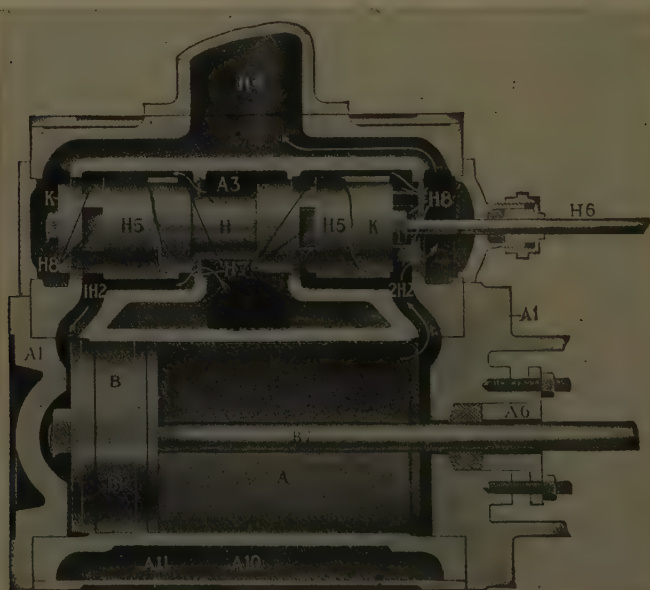
FIG. 85.—PHANTOM VIEW OF CYLINDER AND SLIDE VALVE

grooves around the piston, and spring out tightly against the cylinder surface, thus preventing leakage of steam past the piston. Likewise a stuffing box with packing *A6* is provided where the piston passes through the crank-end cylinder head *A1* in order to prevent steam leakage from the cylinder.

The reciprocating motion of the piston is transformed into rotary motion at the shaft *F* by means of the crosshead *D*, connecting rod *C*, and the crank *F*. The crankshaft *F* is mounted in substantial bearings *F2*. The flywheels, from which belts

may be taken to drive various machines, are carried by the crank-shaft.

Steam distribution and control is accomplished by means of a single valve *H*, or a number of valves depending upon the type of valve mechanism employed as shown in Figs. 83 to 89 inclusive. The valve or valves are placed in a chest or passageways cast integrally with the cylinder or in some cases bolted to the side



(Vacuum Oil Co.)

FIG. 86.—SECTION THROUGH CYLINDER AND PISTON VALVE

or top of the cylinder. The *D*-slide valve *H* of Figs. 83 and 85, is the invention of William Murdock, valued employee and plant superintendent of Boulton and Watt. Watt's early valves were of the poppet type, remotely resembling those of Fig. 89. Murdock's *D*-slide valve was adopted, and largely used by Boulton and Watt in their later engines. The motion of the slide valve and piston valves is one of reciprocation, parallel to the cylinder axis. But the length of valve stroke is usually not more than a tenth of the piston stroke. The position of the valve in its stroke

is always approximately 90 degrees of crank angle ahead of the piston. The valve is actuated by an eccentric *G1*, Fig. 83. Similar to the piston rod, the valve rod enters the valve chest through a stuffing box *H6* which prevents leakage of steam.

Steam enters the steam chest through a steam pipe and is shown entering the head end of the cylinder through port marked *1H2* in Fig. 83, the port being uncovered by the valve at *H*. At the same time the exhaust steam is escaping from the crank end of the cylinder through port *2H2* into the exhaust pipe connection *H4*. Thus the two ends of the cylinder are functioning simultaneously.

Four-valve arrangements are shown in Figs. 87, 88, and 89, there being an independently controlled steam and exhaust valve for each end of the cylinder. By this arrangement, a more perfect control of the steam distribution is obtained with a consequent higher economy of its use.

150. The Corliss Engine. The Corliss valve and valve gear, one form of which is shown in Figs. 87 and 88, was invented by George H. Corliss of Providence, Rhode Island, in 1850. It has been much used as stationary, factory, and mill engines.

The valve consists of a cylindrical bar whose axis extends across the cylinder ends perpendicular to the cylinder axis. Portions of the cylindrical valve are cut away to form a clear passage into the cylinder when the valve is in a certain position. The valve motion is one of oscillation or partial rotation about its axis.

The valves may be operated by two eccentrics on the main shaft, one eccentric for the steam valves and one for the exhaust valves. Valve movements also may be effected by a single eccentric which oscillates a wrist plate as *H10*, shown in Fig. 87. They are then operated through short rods attached to the wrist plate rim at four suitable points. Many variations as to the details of the valve-operating mechanism have been evolved by different inventors.¹

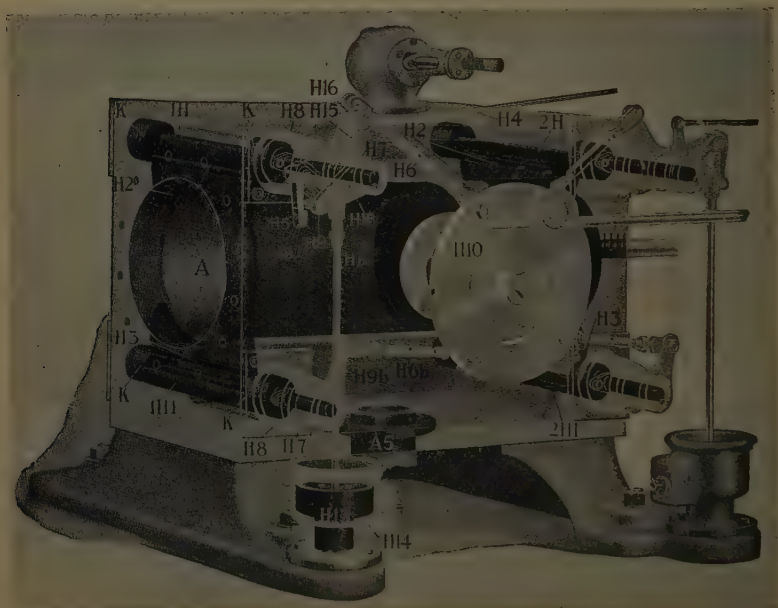
The steam-inlet valves *1H* and *2H*, Fig. 87, are shown sup-

¹ See William Ripper, *Steam Engine Theory and Practice*.

Robert H. Thurston, *A Manual of the Steam Engine*.

Edgar MacNaughton, *Elementary Steam-Power Engineering*.

ported only at their ends by bearing surfaces *K*. The valve stems *H7* are supported in guide brackets *H8* bolted to the cylinder walls. Three arms are assembled upon the valve stem *H7*. The outer arm *H9* is keyed to the valve stem, and connected to the dashpot piston *H13* by the rod *H12*. The small arm *H15* carrying a knock-off cam *H18* is free to rotate on the valve stem *H7* and it is operated by the governor (not shown) through the

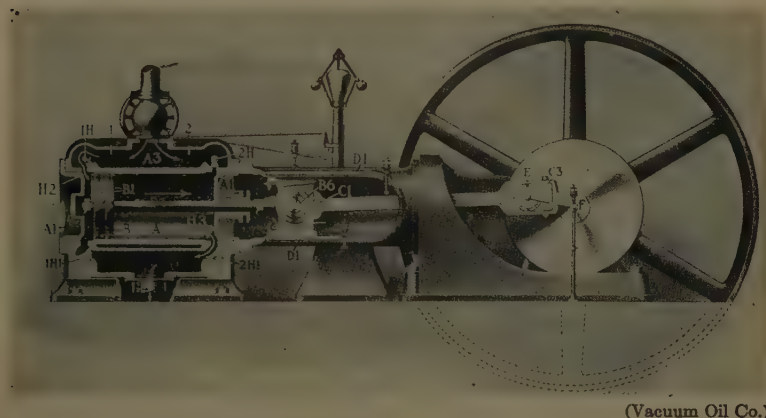


(Vacuum Oil Co.)

FIG. 87.—PHANTOM VIEW OF CORLISS CYLINDER AND VALVES

rod *H4*. The bell crank *H16*, which also rotates independently from the valve stem, carries a pawl *H5* on one arm. The other arm is tied to the wrist plate by the rod *H6*. The pawl is able to engage a latch plate *H17* which is attached to the arm *H9*, thereby opening the valve from the eccentric and wrist-plate movements. At the proper instant in the stroke of the piston, the pawl is automatically disengaged when its rear end strikes the knock-off cam *H18* on the arm *H15*. The exact instant in the forward stroke of the piston when this happens is determined by the governor in accordance with the load demand upon the engine.

When the valve is opened by the engagement of the pawl, the dashpot piston *H13* is raised in the dashpot cylinder by the rod *H12*, producing a partial vacuum in the dashpot. The force produced by the vacuum serves to close the inlet valve *1H* very quickly. Hence the steam-inlet valves are opened by the direct action of the eccentric operating through the wrist-plate rods and arms, disengaged by governor action and rapidly closed by dashpot action.



(Vacuum Oil Co.)

FIG. 88.—VERTICAL SECTION THROUGH THE CYLINDER OF A CORLISS ENGINE

The Corliss valve gear was a decided improvement over previous arrangements because:

1. It gives more accurate control of the steam distribution, producing for all engine loads a pressure-volume diagram closely approaching the ideal by its quick action and the large steam passages made possible.

2. The clearance volume was appreciably reduced by placing the valves close to the ends of the cylinder, thus shortening the steam passages.

3. The four valves, capable of independent adjustment, were important features in improving the *PV*-diagram.

The Corliss valve and valve gear possess two serious disadvantages. It is impossible to operate it at high-rotation speeds because of rapid wear and development of maladjustments. The

usual speed for large units is from 80 to 125 revolutions per minute. A few small engines have been built to operate at 175 or 200 r.p.m., but their performance never was entirely satisfactory. Hence Corliss engines are large, heavy, and costly machines. The second difficulty is met when the attempt is made to use high-pressure and highly-superheated steam (pressures above 175 pounds per square inch and temperatures above 500 degrees F.) The expansion and working of the long cylindrical valves produce serious difficulties due to warping of the valves. Hence the Corliss engine has been replaced in modern power plants by other styles of engine or by steam turbines.

151. The Lentz Poppet-Valve Engine. To meet the demand for the higher engine efficiencies to be gained by the use of higher pressures and superheats, Hugo Lentz developed, about 1900, the Lentz design of poppet valve shown in Figs. 89 and 90.

The steam distribution is controlled by two inlet and two exhaust valves. The valve is a short hollow spool-shaped cylinder of cast iron *2H* (Fig. 89). It is made hollow to permit rapid opening and closing of the steam port by the double-ported valve design. The two ends of the valve seat upon two ground rings *H5* making two steam-tight joints when in the closed position. The valves are opened by a lever working over a fulcrum which is under governor control. The lever is actuated by rods from eccentrics on a lay shaft, best seen in Fig. 90. The double-ported valve feature gives full port opening with a very short valve lift. Springs in the valve bonnet *1H* (Fig. 89) close the valves and hold them in that position. Dashpots in the bonnet cushion the valves gently down on the valve seats, thus preventing rapid deformation of the valve and seats by percussion. The time each valve is held open is determined by the governor-controlled fulcrum, which is pulled from under the valve lever when it is to be closed. Independent valve setting may be obtained for each valve, since each has its own operating eccentric.

The use of steam temperatures of 500 degrees F. and above makes it impossible to use packings of any kind on the valve stems and piston rods because all such packings quickly carbonize, and lubricants quickly carbonize also. Hence valve-stem packings

have been entirely eliminated. The hardened surfaces of the stems are carefully ground to fit the bonnet valve-stem bushings within 0.001 in. From four to eight shallow grooves, a few

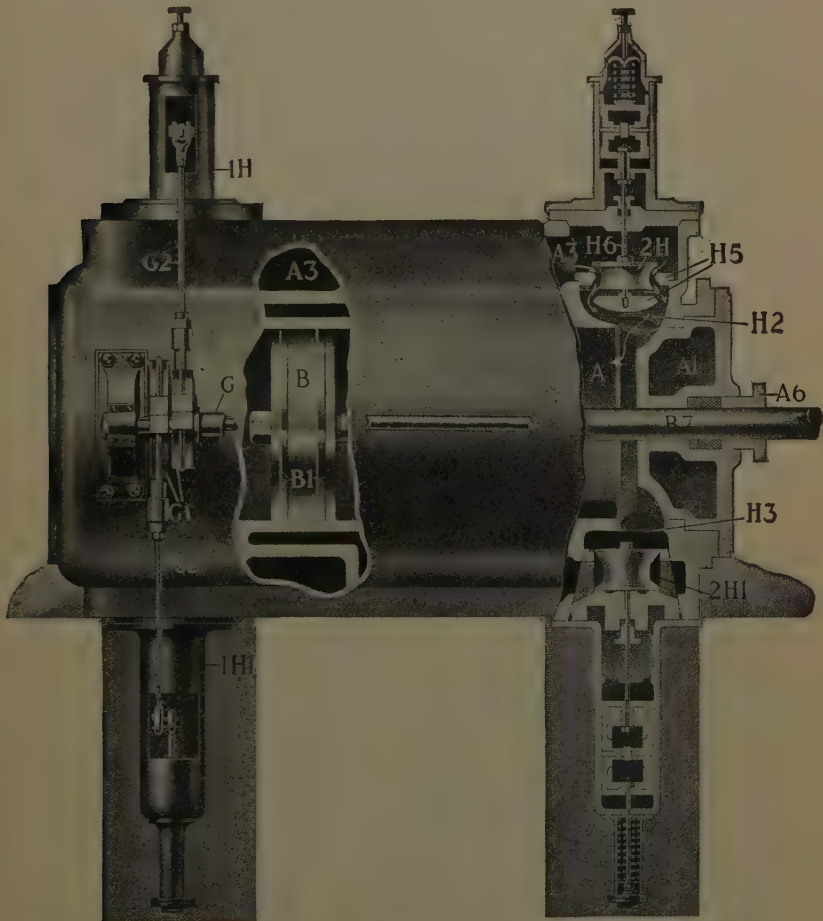
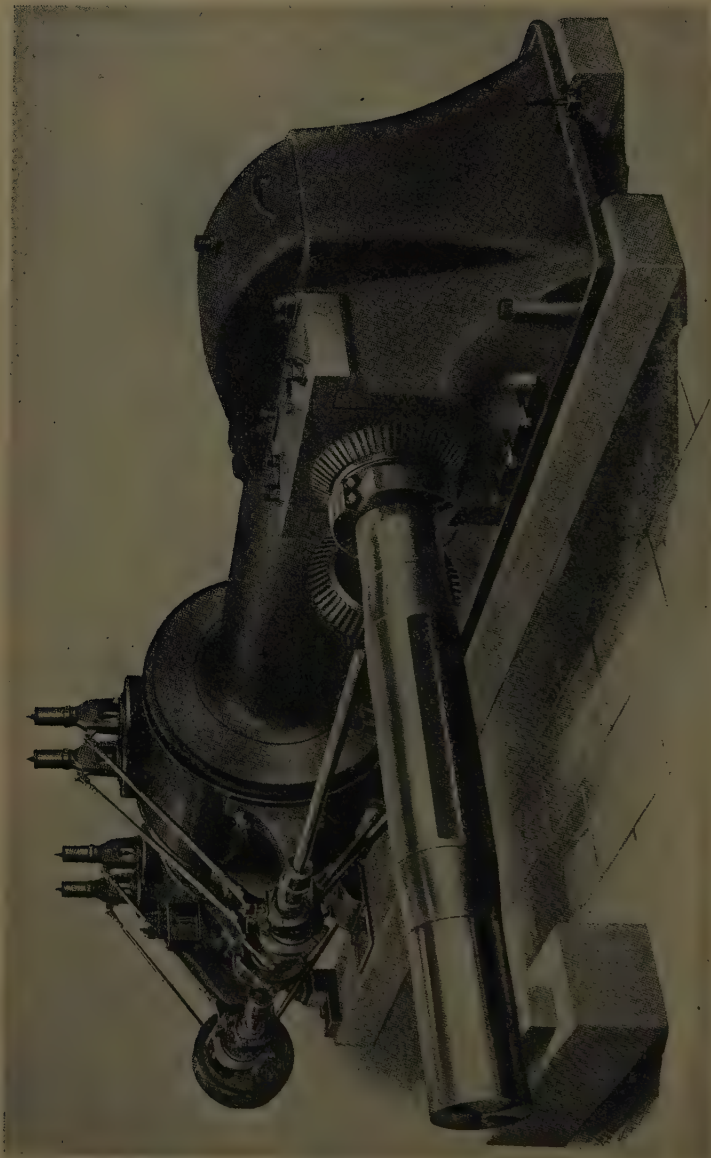


FIG. 89.—CYLINDER AND VALVES OF LENTZ ENGINE

thousandths of an inch deep, are turned in the stems. The close fit and the grooves in the stem serve to make a labyrinth packing which is steam-tight, and which needs no lubricant. The piston rod is packed by a series of metallic rings which clear the rod by about one-thousandth of an inch, thus making a frictionless labyrinth packing similar to the valve-stem packing.



(Erie City Iron Works)

FIG. 90.—LENTZ POPPET-VALVE TANDEM-COMPOUND ENGINE

152. UnafLOW Engines. The credit for having made this type of engine a commercial success is due to Professor Johannes Stumpf of the Technische Hochschule, Berlin.¹ But undoubtedly the credit for the invention of the arrangement together with the first proposal to use high-pressure steam (1000 to 3000 pounds per sq. in.) belongs to Jacob Perkins, who brought out the idea in England about 1820.²

It remained for Professor Stumpf to bring out a correctly designed unafLOW engine at the psychological moment. Consequently he has won much of the credit for the invention. Although Stumpf's engine came into use in Europe in 1908, it was not introduced into the United States until 1912. But since that time it has rapidly won the field formerly held by other types of steam engine and small steam turbines having rated capacities varying from 25 horsepower to 1500 horsepower or thereabouts, because of its superior steam economy, especially at partial loads, and because of its simple, rugged construction. Recently two very large unafLOW engines have been installed to drive steel-mill rolls. Both of these engines are provided with four 36-inch by 60-inch cylinders, one engine being reversible, the other a continuous drive. The continuous-drive engine is rated at 14 000 horsepower with the initial steam pressure at 240 lbs. per. sq. in. gage, and exhausting into a condenser with 26 inches of vacuum. The rated speed is 150 r.p.m. The reversing engine is estimated to be capable of developing momentarily a maximum of 30 000 horsepower.³ UnafLOW engines are also in use on locomotives, hoisting engines, and marine engines.

153. Design Features of UnafLOW Engines. There are three distinctive features which, when combined, have produced the decided improvement in heat efficiency over all older forms of steam engines (see Fig. 91).

1. The exhaust ports are placed around the middle of the

¹ See J. Stumpf, *The UnafLOW Steam-Engine*. Second edition, 1922.

² See Power, p. 718, May 6, 1924; p. 227, Feb. 10, 1925; p. 368, Mar. 10, 1925; p. 451, Mar. 24, 1925; p. 676, Apr. 28, 1925; and p. 412, Mar. 16, 1926.

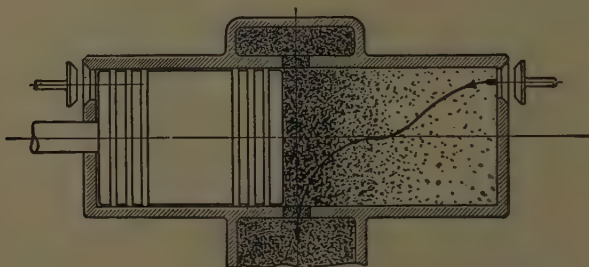
Elijah Galloway, *A History of the Steam Engine*.

R. S. Stuart, *History and Anecdotes of the Steam Engine*.

London Journal of Arts and Sciences, pp. 93, 99, 275, and 285. 1827.

³ See Power, June 22, 1926, p. 968.

cylinder. This eliminates the return flow of the expanded relatively cold and wet steam back over the cylinder surfaces, the steam ports, and valves. This is probably the most important feature of the unafLOW engine in producing high engine-economy. It is a step nearer the realization of the first principle of engine efficiency worked out by Watt, namely, *keep the cylinder as hot as the entering steam.*



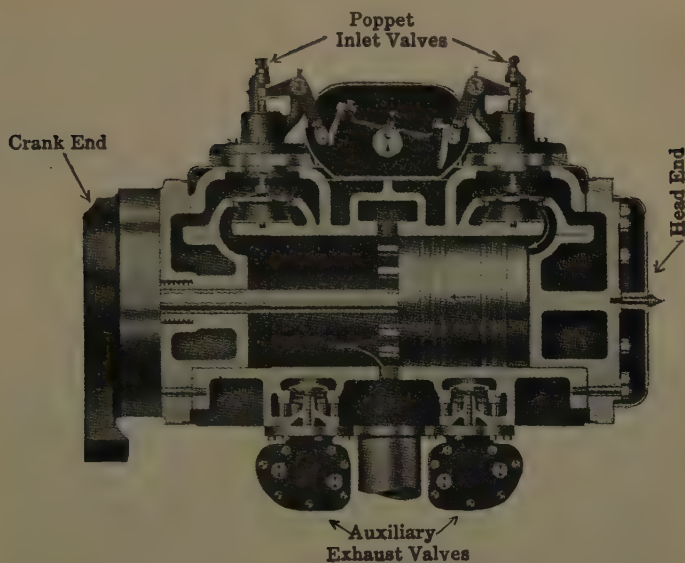
(Nordberg Manufacturing Co.)

FIG. 91.—UNAFLOW CYLINDER

2. The elongated piston is made to serve as the exhaust valve.

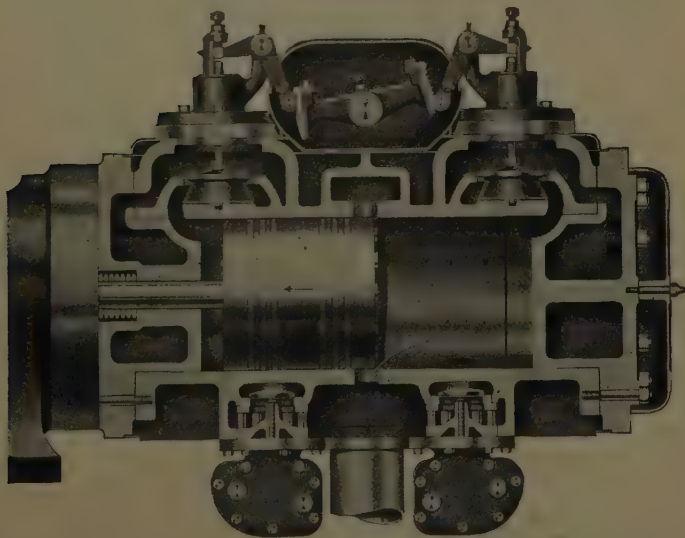
3. The cylinder heads are steam-jacketed to keep them hot and even to impart heat to the compression steam. These points are illustrated in Figs. 91 to 94 inclusive. The steam valves are usually of the Lentz poppet type, although not necessarily so. For some high-speed engines, slide valves or piston valves are used.

In European countries, where the unafLOW engine was first developed, practically every steam engine is operated condensing. Shortly after the piston starts on its exhaust stroke the exhaust ports are covered, thus compressing the trapped low-pressure steam during about 90 per cent of the stroke. But in condensing operation, the pressure of the steam at the end of compression, even with small clearances, does not rise above the initial throttle pressure. But if the attempt is made to operate the engine non-condensing (exhausting to the atmosphere) as happens frequently in the United States, the terminal compression pressure rises much higher unless provision is made in some manner for delaying the beginning of compression, or by increasing the clearance



(Skinner Engine Co.)

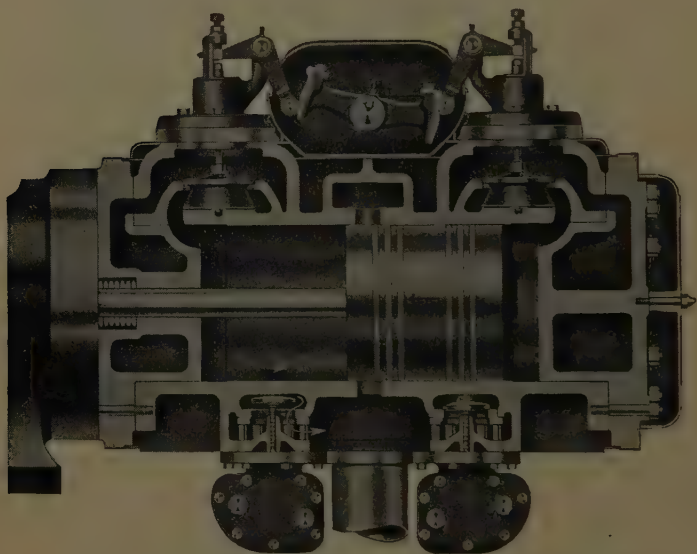
FIG. 92.—CYLINDER OF UNIVERSAL UNAFLOW ENGINE—ADMISSION (BEGINNING IN HEAD END EXHAUST FROM CRANK END)



(Skinner Engine Co.)

FIG. 93.—CYLINDER OF UNIVERSAL UNAFLOW ENGINE (BEGINNING OF EXHAUST FROM HEAD END)

volume into which the steam is compressed. Each of these expedients is used by different engine builders. The functioning of auxiliary exhaust valves, which are operated by the regular valve gear, is controlled by the exhaust pressure of the engine. If the vacuum of an engine should fail suddenly, the auxiliary



(Skinner Engine Co.)

FIG. 94.—CYLINDER OF UNIVERSAL UNAFLOW ENGINE (CRANK END EXHAUST CONTINUING THROUGH AUXILIARY EXHAUST VALVE)

exhaust valves are automatically brought into action or if the engine operates non-condensing, the auxiliary valves operate continuously without hand adjustment to put them in gear as illustrated by Figs. 92 to 94 inclusive.

154. Cylinder Arrangements of Stationary Steam Engines. In Fig. 97 are shown the usual arrangements of horizontal and vertical engine cylinders and frames. Horizontally arranged cylinders are shown in sketches I, II, III, IV, V, VIII, IX, XI, and XIV; while vertical settings or a combination of vertical and horizontal styles are shown in the remainder of the views.

Views I and II show a simple horizontal engine, views VI and VII are for the same engine with its frame and cylinder stood on

end. No. III shows a twin or duplex horizontal engine, both cylinders taking high-pressure steam from the boiler and driving the same crankshaft. The cranks in this arrangement may be 90 degrees or 180 degrees apart. Hoisting engines and steam-shovel engines are usually built on this style. No. IV shows three horizontal cylinders, all taking high-pressure steam and connected to a common crankshaft, whose cranks are set 120 degrees apart. No. V shows a horizontal cross-compound engine, *i.e.* high-pressure steam enters the smaller cylinder, is exhausted after partial expansion into a receiver between the two cylinders, and then flows to the large low-pressure cylinder, where expansion



(Skinner Engine Co.)

FIG. 95.—UNIVERSAL UNAFLOW ENGINE DIRECT-CONNECTED TO GENERATOR
(THIS STYLE IN SIZES FROM 500 TO 1200 HORSEPOWER)

is completed. The cranks are generally set 90 degrees apart. Nos. VIII and IX show a tandem-compound engine. In this style of engine the low-pressure cylinder which is the larger, is usually placed next to the engine frame. No. XI shows a horizontal twin tandem-compound engine. The cranks may be either 90 degrees or 180 degrees apart. No. X shows an angle cross-compound engine in which the high-pressure cylinder is placed horizontally, and the low-pressure cylinder axis is set vertically. Both connecting rods are joined to the same crankpin. No. XII represents the front elevation of a vertical cross-compound engine with the cranks 90 degrees apart; while No. XIII represents a vertical triple-expansion engine with crank angles of 120 degrees. No. XIV shows the arrangement of a

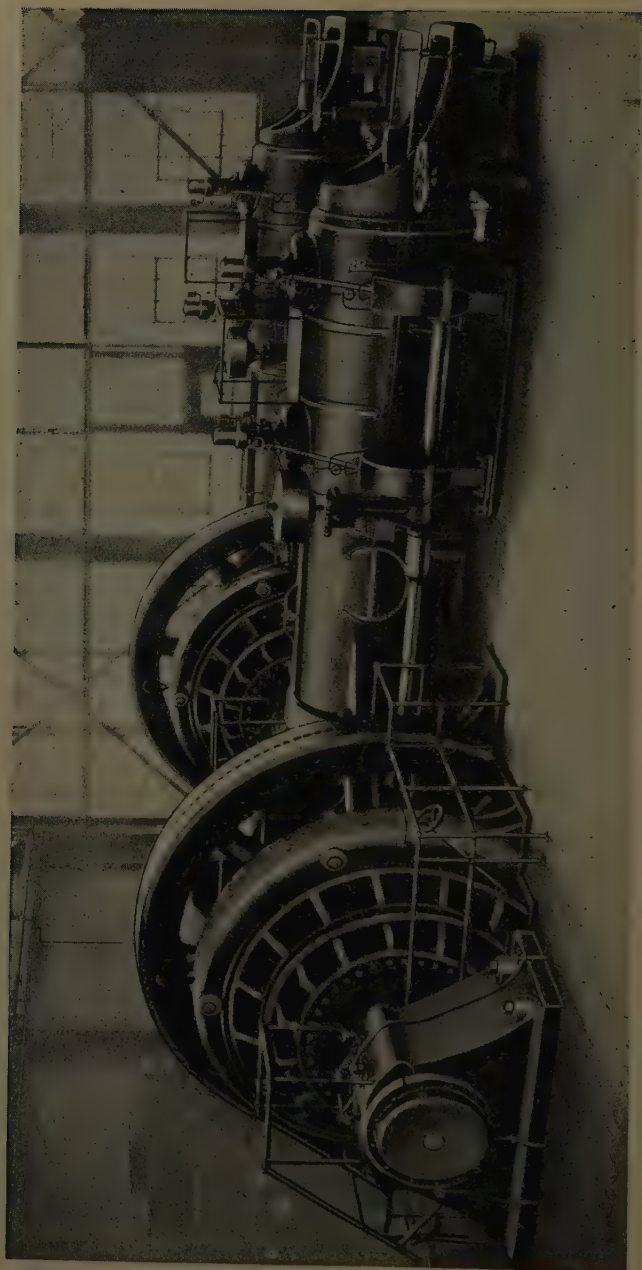
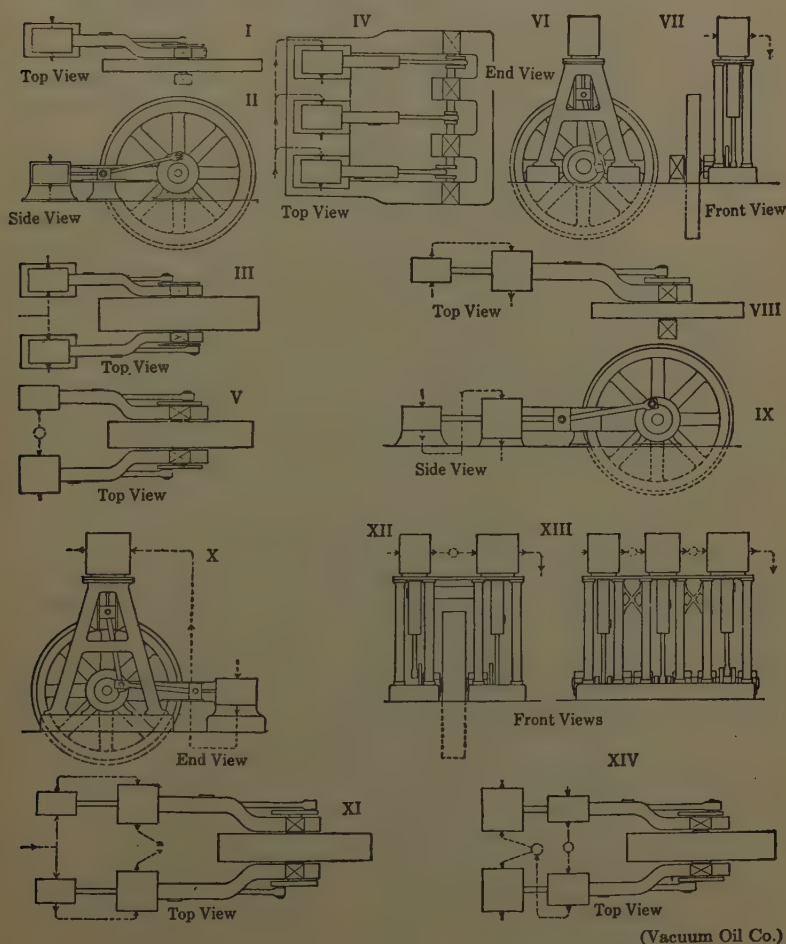


FIG. 96.—Two 36×42 UNAFLOW ENGINES DIRECT-CONNECTED TO 1000 KILOWATT GENERATORS
(Mesta Machine Co.)

four-cylinder triple-expansion horizontal engine with crank-angles 90 degrees apart. There are two low-pressure cylinders, one in tandem with the high-pressure cylinder and the other in tandem with the intermediate cylinder. Practically all engines for marine service, and many land-service engines, are built on the vertical plan because of the greater economy in floor space. The cylinder axes of nearly all stationary, mill-type engines are horizontally disposed. The cylinder axes of unafrow engines may be arranged either horizontally or vertically.



(Vacuum Oil Co.)

FIG. 97.—ARRANGEMENTS OF ENGINE CYLINDERS

155. Locomotive Engine and Boiler.¹ Fig. 98 is reproduced from a photograph of a modern passenger locomotive built for the United States Government during the World War. It is classed as a standard 4-8-2 A mountain-type locomotive. These numbers refer to the number of wheels, there being 4 wheels under the front pony truck, 8 large driving wheels, and 2 small trailing wheels under the fire box. It is equipped with a simple engine on each side whose 27×30 cylinder is provided with piston valves. The piston rods are connected through crosshead and connecting rod to the driving wheels, which are 69 inches in diameter.

The boiler is of the conical wagon-top type, 78 inches in diameter, carrying a working steam pressure of 200 pounds per sq. in. gage. There are 216 $2\frac{1}{4}$ -inch boiler tubes and 40 $5\frac{1}{2}$ -inch tubes. The area of steam and water-heating surface is 4120 square feet, with 957 square feet of superheating surface. The total weight of the engine alone is 327 000 pounds, the total weight of engine and tender is 519 000 pounds. The engine is capable of developing a tractive force or drawbar pull of 53 800 pounds for speeds between zero and six to ten miles per hour. At speeds between 50 and 60 miles per hour, the locomotive is capable of developing 1800 to 2000 horsepower with a tractive force varying from 10 000 to 12 000 pounds. The largest of the modern three-cylinder freight locomotives are capable of developing from 3800 to 4000 horsepower at speeds of 12 to 15 miles per hour.

In the aggregate, the total amount of power obtained from reciprocating steam engines is very large. But in recent years the competition of the internal-combustion engine and the steam turbine have greatly reduced the number of steam-engine installations. The turbine has entirely displaced the piston engine in large central power stations, and very largely replaced it for driving high-speed auxiliaries. In the automotive and aeronautical fields, the internal-combustion engine is the only prime mover of any importance. The modern Diesel engine is rapidly displacing the steam engine for marine and small central station service in which the size of unit varies from 100 to 3000 horsepower. But the steam engine remains superior for service de-

¹ See Marks' *M. E. Handbook*, pp. 1278-1292.



(The Baldwin Locomotive Works)

FIG. 98.—MOUNTAIN-TYPE LOCOMOTIVE

manding variable speeds, low rotative speeds, or heavy starting torques. In many kinds of work, such as driving steel-mill rolls, mine hoists, and in small factories where small, rugged, and reliable prime movers are essential, the steam engine is still the most economical.

156. Watt's Steam Engine-Indicator. In order to measure the varying force of steam against the piston during a cycle, James Watt invented, about the year 1800, a miniature cylinder and

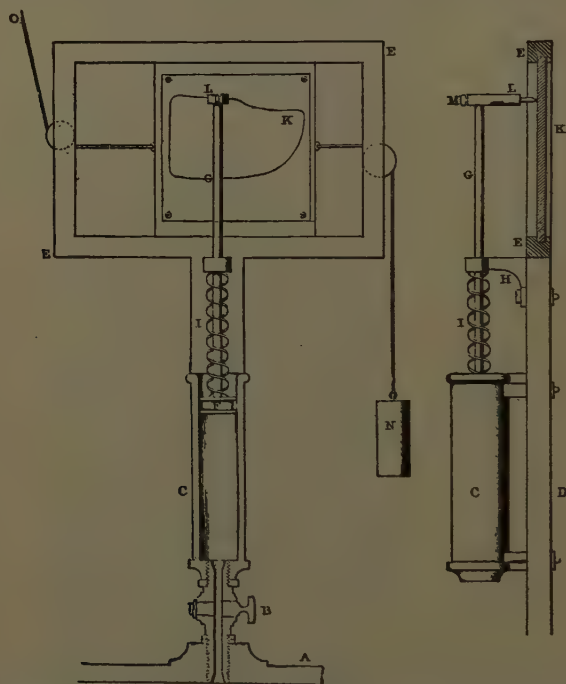


FIG. 99.—WATT'S INDICATOR (From *The Steam Engine*, by Thomas Tredgold)

spring-loaded piston which may be connected to the end of the steam-engine cylinder. One of the early forms of the indicator is illustrated in Fig. 99. It is supposed that Watt invented the spring-loaded piston and its cylinder, but the tracer and sliding board were afterward added by a Mr. Southern.¹

¹ See Thomas Tredgold, *The Steam Engine*, 1838, p. 269.

The early Watt indicator "consisted of a cylinder *C* about $1\frac{3}{4}$ " in diameter, and 8" long, exceedingly truly bored, with a solid piston *F* accurately fitted to it, so as to slide easy by the help of some oil; the stem of the piston is guided in the direction of the axis of the cylinder, so that it may not be subject to jam or cause friction in any part of its motion. The bottom of the cylinder has a cock, *B*, and small pipe joined to it; a flat pillar *D* is screwed to the cylinder of the indicator *C*, and supporting the frame *EE*, which is twelve inches by seven inches with the upper and under rail grooved to retain the sliding board *K*.

"The piston rod *G* is about five-eighths of an inch in diameter, and sixteen inches long; and *H* is the guide for it screwed to the pillar *D*, at about six inches above the top of the cylinder *C*.

"A spiral spring *I* is attached to the piston at *F*, and to the guide at *H*. It should be about seven inches long when at rest, and of such a strength as to allow the piston *F* to descend to within about an inch of the bottom of the cylinder *C*, when it is loaded with fifteen pounds upon every square inch of its area; and the spring should admit of being compressed one inch and a half.

"The board or pannel *K* slides in the grooves of the frame *EE*, and should be seven inches square; and the small brass slider *L* should be set at any height on the piston rod *G*, by means of a screw. A short pencil is inserted in the other end, with a weak spring to push it against the surface of the board *K*, which is caused to slide by a weight *N*, attached to a line passing over a pulley; the opposite line *O* being attached to any convenient part of the parallel motion of the engine, so as to cause the board *K* to traverse a space of about fourteen inches and a half during each half stroke (half revolution is evidently meant) of the engine." ¹

The movements of the spring-controlled piston produced the ordinates and the movements of the sliding board produced the abscissas of the pressure-volume diagram. The *PV*-diagram thus produced by an indicator is generally known as an *indicator card*. The full movement of an engine piston having a four-foot

¹ Quoted from Thomas Tredgold, *The Steam Engine*, 1827, p. 276.

For views of Indicators in use about 1840, see Power, p. 948, Sept. 21, 1926; and p. 247, Aug. 16, 1927.

or an eight-foot stroke obviously could not be recorded full size upon the seven-inch board; hence a reducing motion which would exactly reproduce the piston movements on a reduced scale, proportionally correct at every point in the stroke is another invention made by Watt. Various forms of reducing motions, four of which are illustrated in Figs. 110 and 111, are in use.

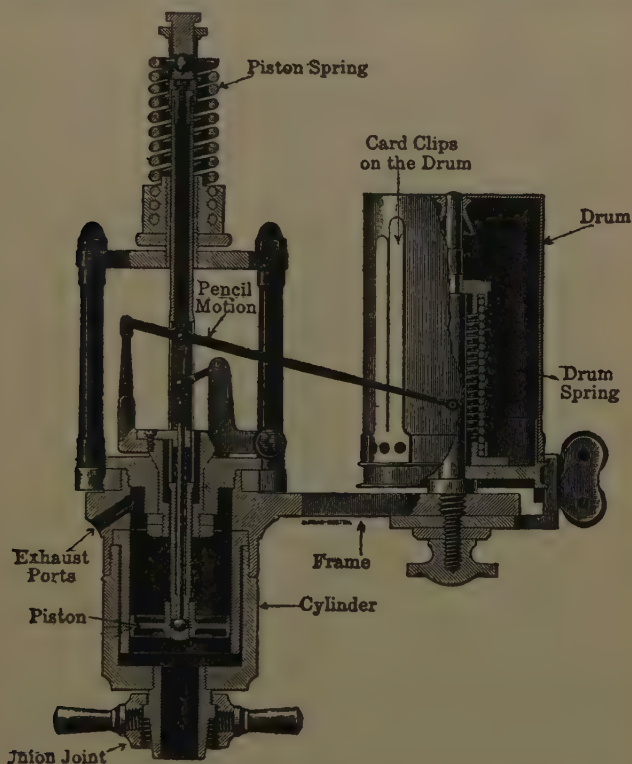
It should be stated that the dimensions of Watt's cylinders to which he applied his indicator varied from 30 inches in diameter with a four-foot stroke to 90 inches in diameter with an eight-foot stroke. Previous to 1825, the steam pressure employed in Boulton and Watt engines never exceeded 25 pounds per square inch. The usual pressures varied between five pounds and about fifteen pounds per square inch above that of the atmosphere. These very large low-pressure engines were operated at speeds which would correspond to 5 or 10 revolutions per minute for modern engines.

For the steam pressures and engine speeds in use at that time, Watt's indicator was entirely adequate. But to cope with the growing pressures and increasing speeds, the indicator has necessarily passed through several evolutionary stages.

157. Modern Indicators. Forms of modern indicators and equipment are shown in Figs. 100 to 107 inclusive. Modern design and manufacturing methods have developed rugged instruments of refinement which produce results having a high degree of precision.

Fig. 100 is a sectionalized view of a modern outside-spring steam-engine indicator. In principle, it embodies the main features contained in Watt's first indicator, but it differs in detail of design. It consists of a cylinder in which a piston is moved by the steam pressure in the engine cylinder. A suitable spring attached to a long piston rod restrains the movements of the piston proportionally to the pressure acting on the piston. The piston movement is multiplied about six times by the recording stylus. A light cylindrical drum, rotatably mounted on an axis parallel to the axis of the indicator cylinder, carries the card upon which the stylus leaves its record. The move-

ments of the piston produce the pressure components and the rotation of the drum generates the volume components of the *PV*-diagram recorded on the card. The multiplying pencil motion consists of a mechanism whose kinematic principle is that of the pantograph. This as well as other straight-line motions (see Figs. 100 to 107 inclusive) give the pencil point a movement



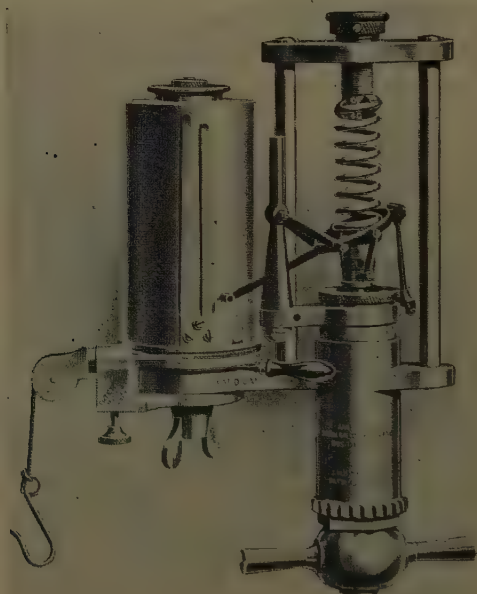
(Crosby Steam Gage and Valve Co.)

FIG. 100.—MODERN OUTSIDE-SPRING INDICATOR

exactly parallel to that of the indicator piston as well as parallel to the axis of the drum. The drum is rotated in synchronism with the movements of the engine piston but through a proportionally reduced distance by means of an indicator reducing rig (Figs. 110 and 111). A strong unstretchable cord is wrapped around the indicator drum. The cord is attached to the reducing rig. The

outward movement of the engine piston and crosshead is transmitted through the reducing rig to rotate the drum through approximately 300 degrees. Upon the return stroke of the engine piston, the drum reverses and is returned to the initial position by a long helical drum spring, thus completing the cycle.

A set of carefully calibrated indicator springs are supplied with each instrument. From 10 to 15 springs are usual in a com-

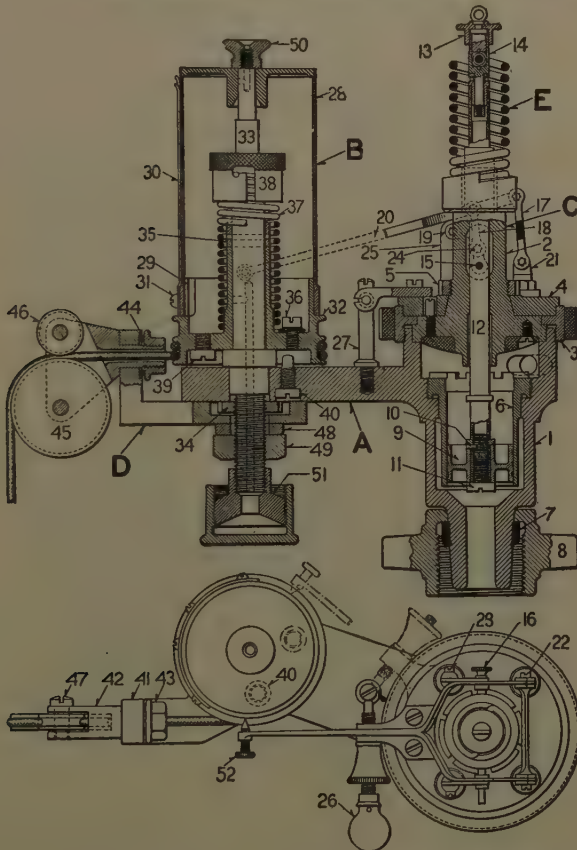


(American Steam Gage & Valve Mfg. Co.)

FIG. 101.—AMERICAN-THOMPSON OUTSIDE-SPRING INDICATOR

plete set. Springs numbered 8, 12, 20, 30, 40, 60, 80, 100, 150, and 200 are generally included in the set. The number of the spring indicates that the relations between the area of the indicator piston, the strength of the spring, and the pencil multiplying motion are such as to cause the stylus point to be raised exactly one inch when the steam pressure in pounds per square inch beneath the indicator piston equals the spring number. Thus a number 80 spring will permit the stylus point to make a vertical record exactly one inch long on a card wrapped around the drum;

twenty pounds pressure would then raise the stylus $\frac{1}{4}$ inch, and so on in proportion to other pressures. Indicator-piston areas are generally made 1, $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{10}$, or $\frac{1}{40}$ square inch. Several pistons of different diameters with corresponding sleeves are provided with some indicators, thus giving a choice of a wide range

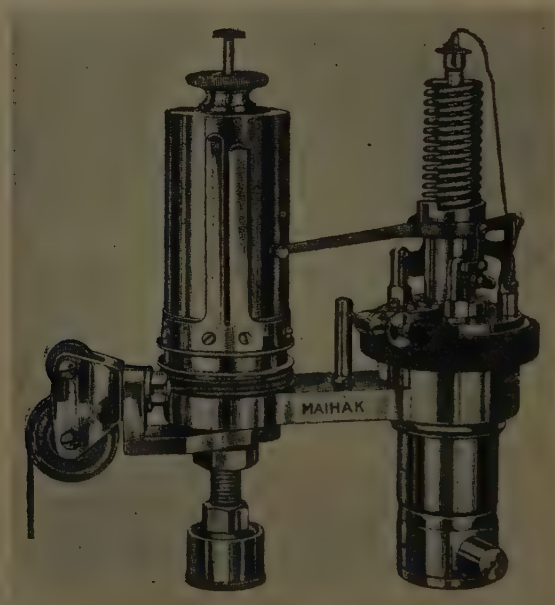


(Bacharach Industrial Instrument Co.)

FIG. 102.—THE MAIHAK-TYPE INDICATOR

of spring scales with a small number of springs. For steam-engine work, piston areas of 1 inch and $\frac{1}{2}$ inch are generally used. For internal-combustion engines and ordinance work, the smaller pistons are used. Indicator drums may be 1, $1\frac{1}{2}$, or 2 inches in diameter, and even smaller for very high speed work.

Fig 101 shows another form of steam-engine indicator which, fundamentally, is no different from the one shown and described above. There is however, some difference in the form and design of the details, the principal difference being in the arrangement of the linkages which constitute the pencil-multiplying motion. This form of instrument was developed in the period between 1860 and 1900. With the arrival of the explosion engine in which the forces are greater and are applied much more rapidly, and the rotative speeds are from five to ten times higher than those for steam engines, the older designs of indicator are not satisfactory.

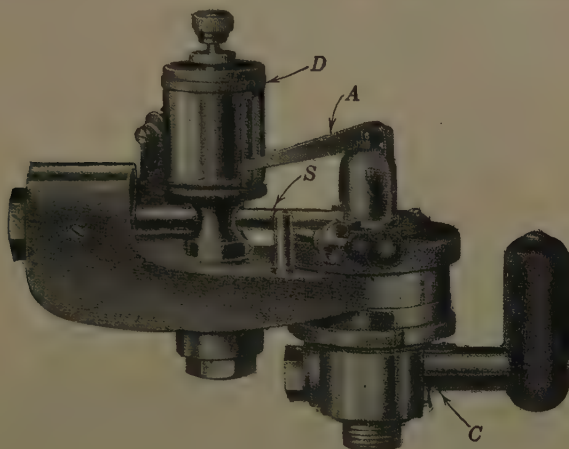


(Bacharach Industrial Instrument Co.)

FIG. 103.—MAIHAK No. 2 INDICATOR

In Figs. 102 and 103 is shown one style of the *Maihak indicator* which is essentially the same in design as those mentioned above. But some refinements such as lighter drum, lighter but more rigid pencil motion, and more rugged construction throughout, are incorporated. These improvements reduce distortions in the indicator diagram due to inertia of the moving parts to sufficiently small values at engine speeds as high as 500 r.p.m.

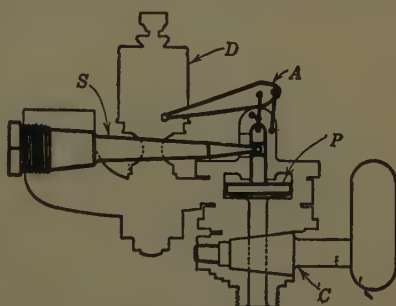
For engine speeds up to the highest, namely, 2000 to 8000 r.p.m., no indicator that is satisfactory in every respect is yet available. The form of the Maihak indicator shown in Figs. 104 and 105 is very good for speeds up to 2200 r.p.m. and for pressures up to



(Bacharach Industrial Instrument Co.)

FIG. 104.—MAIHAK INDICATOR FOR SPEEDS UP TO 2200 R.P.M.

1400 pounds per square inch. The indicator-card drum *D* for this instrument is about one inch in diameter. It is operated by a steel tape attached to a suitable reducing gear. The pressure spring has been changed to a round, tapered cantilever which is the most distinguishing feature of this indicator. In this form, the inertia effects can be reduced much below what is possible with the helical form of spring. The natural period of vibration is very high, and the piston rod can be made very short and integral with the piston. This design permits the entire indicator structure to be much more rigid. The Maihak

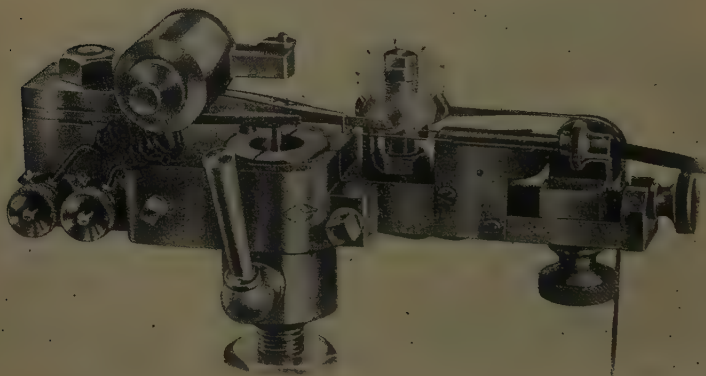


(Bacharach Industrial Instrument Co.)

FIG. 105.—SKETCH SHOWING ARRANGEMENT OF MAIHAK HIGH-SPEED INDICATOR

high-speed indicator produces a card about one inch high and two inches long, which is large enough for a good degree of accuracy in determining engine horsepowers.

The Collins micro-indicator,¹ designed by W. G. Collins of the Cambridge and Paul Instrument Company, London, England, is illustrated in Figs. 106 and 107. A small cylinder and a piston controlled by a flat cantilever spring are employed to produce the vertical component of the *PV*-diagram. The cylinder *W* (Fig. 107) and the piston are made of hard stainless steel. Liners



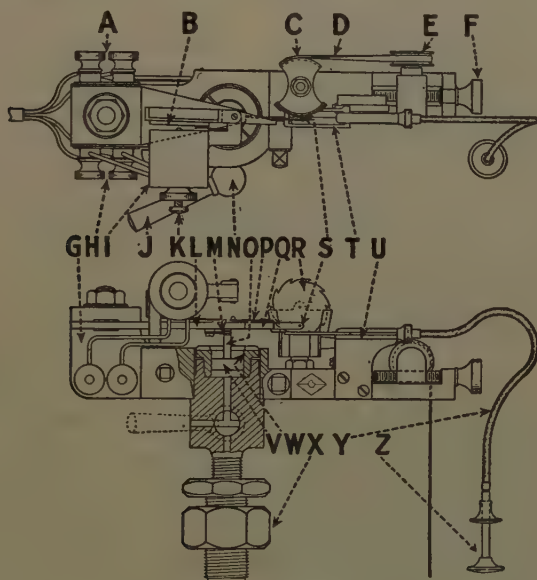
(Cambridge Instrument Co. of America, Inc.)

FIG. 106.—COLLINS MICRO-INDICATOR

and smaller pistons may be placed in the indicator cylinder, thus making the instrument available for measuring high pressures. The short piston rod *O* is formed integral with the piston. It is held in close contact with the pressure spring *L* by means of an auxiliary spring *M*, the spring *L* being firmly bolted to the indicator frame *G*. The recording stylus *S* is attached directly to the pressure spring and is a very light extension of it; hence there is no multiplying pencil motion. The stylus scratches records near the edge of a celluloid disc *R*, which is carried on, and bent to conform to the cylindrical surface of the vertical drum *C* by means of a central pin and clips on the drum. The drum *C* is

¹ See *Engineering* (London), June 9, 1922.

given a reciprocating motion about its axis by the light steel tape *D*, which, after passing over an adjustable guide pulley *E*, is attached to a connecting rod and crank or other suitable reducing motion synchronizing the drum and engine-piston movements. A series of ten diagrams may be taken around the circumference of the celluloid disc, which is about one inch in diameter. The



(Cambridge Instrument Co. of America, Inc.)

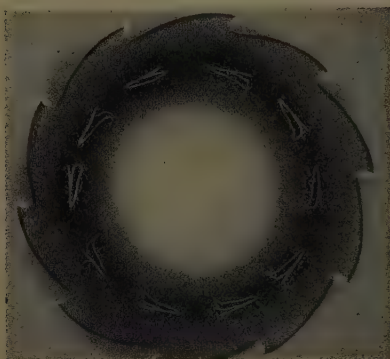
FIG. 107.—COLLINS MICRO-INDICATOR

stylus *S* is brought into contact with the celluloid by an electro-magnet *I*. When the magnet is energized it brings the stylus point against the celluloid with the necessary pressure which can be controlled by a knurled screw *K*. The electric current to the magnet is controlled by an automatic switch which breaks the circuit as soon as one engine cycle is completed.

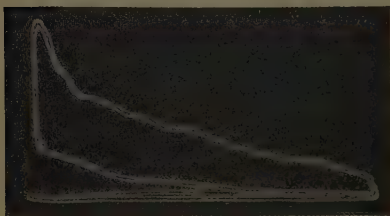
Micro-diagrams, enlarged several times, are shown in Fig. 108. The original record is about 3 millimeters long and about 2.5 millimeters high (about 0.12 by 0.10 inches). The diagrams may be examined at once by means of a suitable small microscope which is provided with the instrument, or the diagrams may be enlarged

to suitable dimensions by photographic methods as shown by Fig. 108 b. The micro-indicator may be used for taking diagrams at speeds up to 3000 r.p.m.

Several forms of *manograph indicators* for high-speed internal-combustion engines have been developed. A corrugated, flexible pressure disc, one side of which is exposed to the fluctuating pressures in the engine cylinder, is employed. These pressures may be balanced and measured by applying, from an external source, controlled and known air pressures. The disc is allowed a small deflection (a few thousandths of an inch). The instant at which the pressures on the two sides of the disc are just balanced is indicated electrically. A timing device indicates the points in the cycle (two of them in every cycle) at which the pressures are balanced. The third part of the instrument consists of the measuring and recording apparatus. A number of forms of manograph indicator, varied as to details, have been developed.¹ Some of them are capable of giving satisfactory results for engine speeds up to 5000 r.p.m.



(a)



(b)

(Cambridge Instrument Co. of America, Inc.)

FIG. 108.—COLLINS MICRO-DIAGRAMS

¹ H. C. Dickinson and F. B. Newell, *A high-speed pressure indicator of the balanced diaphragm type*, Report No. 107, Natl. Advisory Committee for Aeronautics, 1921.

F. W. Burstall, *New form of optical indicator*, Proceedings, Institution of Mechanical Engineers, Jan., 1923, pp. 111-125.

W. J. Stern and H. Moss, *Improved optical indicator*, Aeronautical Research Committee (Great Britain), *Reports and Memoranda No. 925*, Oct., 1924; and Proceedings, Institution of Mechanical Engineers, Jan., 1925, pp. 1-8.

Gale, *Indicator for high-speed engines*, Engineering, March 12, 1926, p. 350.

H. M. Jacklin, *High-speed indicator system*, Journal of the Society of Automotive Engineers, June, 1926; and Mechanical Engineering, May, 1927, p. 545.

Indicators are attached to the engine cylinder by means of a ground union joint *B* (Fig. 100), being connected to a threaded plug cock (Fig. 109). The lower end (Fig. 109 *b*) is screwed into the clearance space of the engine cylinder. The cap on top of the plug cock is removed and the indicator union nut *B* (Fig. 100) screwed on, making a steam-tight joint.

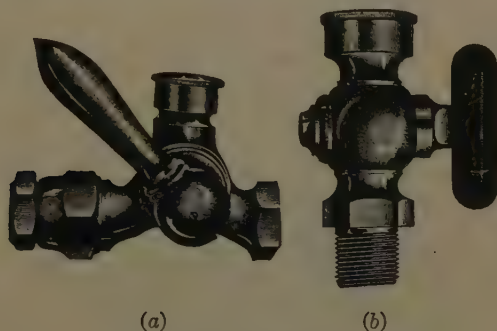


FIG. 109.—INDICATOR COCKS

158. Reducing Motions. The pencil mechanism of an indicator multiplies the piston movements, but the stroke of the crosshead must generally be reduced before being transferred to the indicator drum. Strict proportionality between the engine piston and drum movements must be preserved at every point in the engine and indicator-drum strokes if areas on the indicator card are to represent accurately the foot-pounds of work done by the engine piston.

The *pendulum type of reducing motions* which are illustrated in (a) and (b) of Fig. 110 are more frequently used than any other. They are simple in design with few moving parts, and when accurately constructed give sufficiently accurate reproductions of the piston movements on a reduced scale.

The pendulum *A* of Fig. 110 (a) swings about the fixed pin joint at *G*, located directly above the mid-position of the crosshead. The bottom end of the pendulum carries a fixed pin at *B*. A short link *C*, having a perpendicular slot in which the pin *B* slides, is fixedly attached to the crosshead. The indicator cord passes over a guide pulley *D* and is attached to the pendulum at *H*. The section of indicator cord *HD* should be parallel to the crosshead guides and the guide pulley should be as far from the pendulum as conveniently possible. The relation between the length of drum and piston stroke

is expressed by the following proportion:

$$(341) \quad \frac{\text{length of drum stroke}}{\text{length of piston stroke}} = \frac{GH}{GB}.$$

The length of the pendulum should, if possible, be at least $1\frac{1}{2}$ times the length of piston stroke. When accurately constructed this form of reducing motion will give a very nearly perfect indicator card so far as the volume factor is concerned.

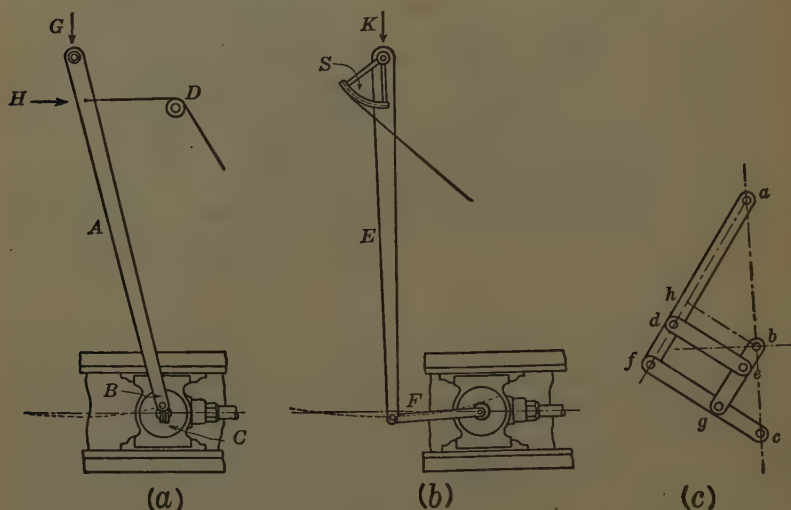


FIG. 110.—REDUCING MOTIONS

The pendulum and link motion shown in Fig. 110 (b) is known as the *brombo pulley*. It differs from the simple pendulum, Fig. 110 (a) by the addition of a link *F* at its lower end and a sector *S* to carry the indicator cord. The link *F* should be approximately one-third the length of the engine stroke. The pin *K* about which the fixed end of the pendulum swings should be so located that the pendulum arm *E* occupies the vertical position when the crosshead is at the mid-point of its stroke. The length of the pendulum arm should be not less than $1\frac{1}{2}$ times the engine stroke; twice the stroke would be preferable. The brombo pulley however, gives a more nearly correct reduction if the sector *S* is not employed, but if instead the cord is attached directly to the pendulum arm, as shown in Fig. 110 (a). Indicator reducing motions of the pendulum type can never give absolutely correct reductions, but the two forms when constructed without a sector and as

described above will give indicator cards whose areas are in error by less than 1.0 per cent.¹ Arrangements of gears and wheels, one form of which is shown in Fig. 111, are also used to some extent, but they cannot be used on short-stroke high-speed engines with a satisfactory degree of accuracy.²

A convenient and usable form of *pantograph reducing motion* is shown in Fig. 110 (c). It consists of four bars connected by pin joints, all free to move. But care has to be exercised in making all joints fit snugly in order not to introduce distortion into the indicator diagram by lost motion in the joints. The pin *C* is fixedly attached

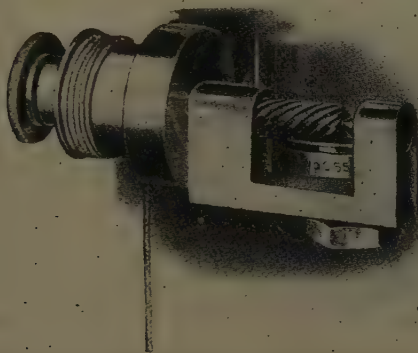


FIG. 111.—HOUGHTALING SPIRAL—GEAR REDUCING MOTION

to the frame of the engine. The end *a* is attached to the engine cross-head and the indicator cord is attached at *b*. The center of the pins *a*, *b*, and *c* must all lie on the same straight line; the opposite links must form a true parallelogram; and the indicator cord must move parallel to the crosshead if a correct reduction is to be obtained. Theoretically the pantograph gives a perfect reduction, but in order to realize perfect results in practice it must be correctly designed and accurately built.³

¹ See E. S. Libby, *Errors of indicator reducing rigs*, Power, Aug. 8, 1911, p. 206.

² See J. C. Smallwood, *Limitations of reducing wheels*, Power, Sept. 24, 1912, p. 445.

³ For more complete information on straight line motions and reducing motions, see:

Julius Weisbach, *Mechanics of Machinery*.

Wm. J. Rankine, *Machinery and Millwork*.

D. A. Low, *Heat Engines*.

159. **The Steam-Engine Cycle and Indicator Diagram.** A diagrammatic sketch of a simple slide-valve engine cylinder and the indicator diagram for the head end are shown in Fig. 112. The diagram is located vertically below the cylinder. The scale of abscissas in Fig 112 (b) corresponds to the changing volume between the left cylinder head and the piston in its successive positions throughout the cycle, the clearance volume being included.¹ The value of the clearance volume of an engine may be

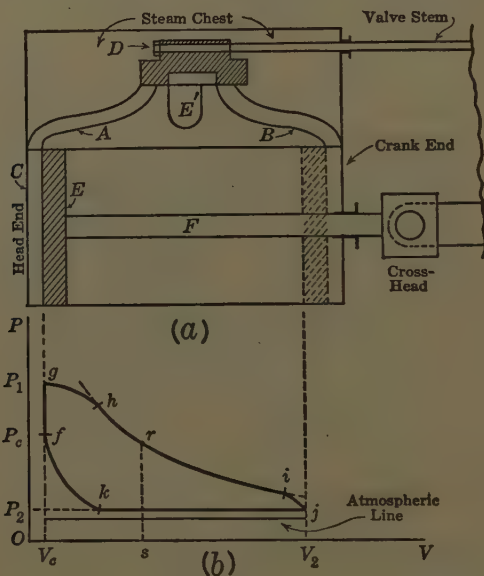


FIG. 112. THE STEAM-ENGINE INDICATOR DIAGRAM

measured by filling it with a known volume of water, or it may be determined from the engine drawings and specifications by computation. It is represented in Fig. 112 (b) by the distance OV_c along the V -axis.

The engine cycle begins with the *admission of steam* through port A just as the piston reaches its extreme head-end position, and is about to begin a new forward stroke. This is indicated by the point f. The pressure in the cylinder immediately rises nearly to boiler pressure, as indicated at g. As the piston moves for-

¹ See definition of clearance volume, page 89, herein.

ward, the steam port remains open, and steam admission continues, as indicated by the line gh , which shows the pressure and volume changes in the head end of the cylinder. The admission line of the indicator diagram droops for two reasons. The port area is not large enough to admit steam fast enough to maintain the pressure in the rapidly growing volume behind the piston, and very shortly the valve begins to close the port. Since the valve velocity is not infinitely great, port closure is not instantaneous. At the point h , the port is just completely closed. This is known as the *point of cut-off*. From this point to near the end of the stroke at i , steam expands nearly isothermally from h to i . At point i , the valve opens port A to the exhaust passage E' , permitting the steam to escape into the condenser or to the atmosphere. Exhaust continues during a large portion of the back stroke of the piston. At point k , the exhaust port is closed, and the steam remaining in the cylinder is trapped and compressed along the isothermal curve kf . At f steam is again admitted from the boiler, and the cycle repeated indefinitely as long as the engine is in operation.

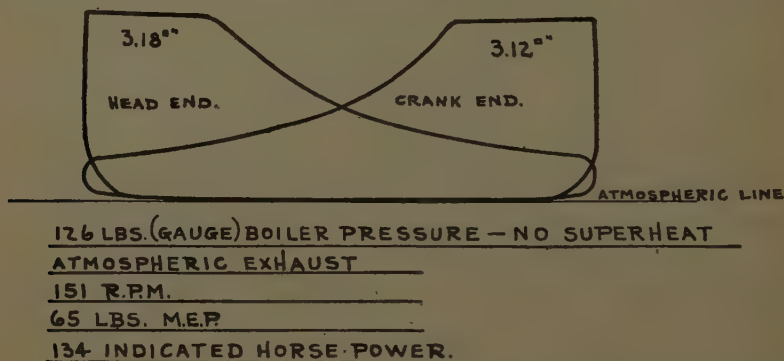
The area V_cghijV_2 represents, to scale, the work done on the face of the piston by the steam during the forward stroke. The area $V_2j k f V_c$ represents the work done on the steam by the piston in pushing the steam from the cylinders and compressing that caught in the cylinder at the end of the exhaust period. Therefore, the net effective work done on the left face of the piston during the two strokes of a complete revolution of the engine shaft is represented by the area $fghijk$.

Meanwhile, a corresponding series of events has been taking place in the crank end of the cylinder in case the engine is double acting. If both indicator diagrams are placed upon the same card, their appearance would be as shown in Fig. 113.

There are four events which occur during an engine cycle, namely, *admission*, *cut-off*, *release*, and *compression*. These events divide the cycle into four corresponding periods. Because the valve does not open and close the ports instantly, it is sometimes difficult to determine, by inspection, the exact location of the beginning of cut-off and release on the diagram. This may be

done sufficiently accurately, however, by extending the expansion line, isothermally, both ways as shown in Fig. 112 (b). The points *h* and *i* at which the extended curve first deviates noticeably from the record made by the indicator stylus are taken as the correct locations of the two events. The same procedure may be followed in locating the exact points at which compression and admission begin.

The appearance of an indicator card as it is taken off the indicator drum may be seen in Figs. 113, 121, and 122. The card



(Murray Iron Works Co.)

FIG. 113.—INDICATOR CARDS FROM 12 X 24 MURRAY CORLISS ENGINE

carries only the pencil record forming a closed area, and the atmospheric line which is drawn after the diagram is taken. After the diagram is recorded, the indicator cock is closed, thereby releasing the steam confined below the piston to the atmosphere through a small port in the cock. The upper part of the indicator cylinder is at all times in communication with the atmosphere through ports (Fig. 100). Hence a line drawn by the indicator pencil, when the indicator cock is closed, gives the location of the atmospheric pressure as a datum from which all pressures on the card may be measured. Knowing the scale of the indicator spring used, it is then possible to locate the level of the *V*-axis at the level of absolute zero pressure. From the ratio of the indicator-drum stroke to the engine stroke, the area of the engine piston, and the value of the clearance volume, the position of the *P*-axis beyond the end of the diagram is readily located. The

intersection of the P -axis and the V -axis at O , Fig. 112 (b), then locates the origin of the expansion and compression curves. At any point r on the boundary lines of the diagram, the instantaneous absolute pressure of the steam in the cylinder is represented, to scale, by the ordinate rs . The total volume of steam in the cylinder at the same instant is represented by the abscissa Os .

160. Hypothetical Indicator Diagrams with and without Clearance. In designing steam engines and studying their performance, it is convenient to employ conventionalized diagrams, called hypothetical diagrams (Figs. 114 and 115). In construct-

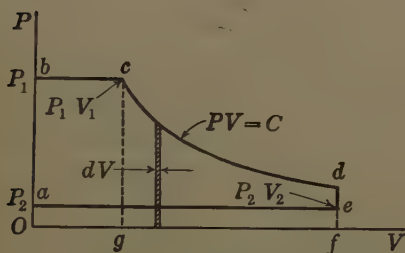


FIG. 114.—HYPOTHETICAL DIAGRAM WITHOUT CLEARANCE

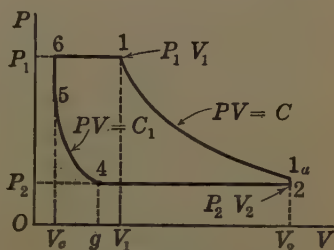


FIG. 115.—HYPOTHETICAL DIAGRAM WITH CLEARANCE

ing such diagrams certain assumptions, not realizable in practice, are made. For Fig. 114, the assumptions are: (1) no clearance volume; (2) full boiler pressure up to cut-off; (3) instantaneous valve action; (4) isothermal expansion; (5) release before the pressure reaches that of the exhaust by expansion; and (6) no compression. (The exhaust pressure in non-condensing operation is assumed to be that of the atmosphere, and in condensing operation it is assumed to be the pressure in the condenser. See Figs. 119 and 120.)

In the hypothetical diagram with clearance, Fig. 115, the assumptions are: (1) normal amount of clearance; (2) full boiler pressure up to cut-off; (3) instantaneous valve action; (4) isothermal expansion and compression; and (5) release occurs before the pressure falls to that of the exhaust.

161. Construction of Isothermal Curves. It is shown by geometry that the isothermal curve is a rectangular hyperbola. A

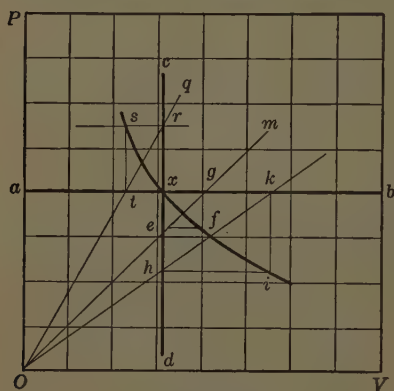


FIG. 116.—CONSTRUCTION FOR ISOTHERMAL CURVE.

simple graphical construction is then available for drawing an isothermal through a given point and having a given origin as point *x*, Fig. 116. Draw the two lines, *ab* and *cd*, parallel to the respective *V*- and *P*-axes through the point *x*. Then draw a series of lines radiating from the origin as *og*, *om*, and *ok*. Upon the intercepts of those radials as diagonals construct rectangles as *efgx*, *hikx*, and

rstx. The corners of the rectangles at *s*, *f*, and *i* are points on the required isothermal.¹

162. Expansion Curves for Steam. In the ideal case the expansion and compression curves of an indicator diagram are adiabatics whose equations have the general form $PV^k = K$. But for steam, the value of *k* is far from being a constant. It is different for wet, dry, and superheated steam, and it also varies with the pressure and the degree of superheat. For wet steam whose quality *x* lies between 0.70 and 1.00, the value of *k* is expressed, approximately, by the following equation:

$$(342) \quad k = 1.035 + 0.1x.$$

In unaf flow engines, the cylinder loss due to steam condensation and radiation is less than in contraflow engines. With dry or slightly superheated steam the expansion curve for unaf flow engines is represented very closely by the equation,

$$(343) \quad PV^{1.1} = K.$$

With steam superheated from 50° F. to 200° F. in unaf flow engines,

¹ See Wm. Kent, *Mechanical Engineers' Handbook*, tenth edition, p. 647, for a graphical method of constructing any polytropic.

the exponent in (343) becomes approximately 1.03; and for the same range of superheat in contraflow engines, the expansion curve is represented approximately by (343) when the value of the exponent is 1.06.¹ However, all these curves approach the isothermal so closely that it is difficult if not impossible to distinguish between them in any actual case. Hence it is general practice to assume all expansion and compression curves for actual and hypothetical diagrams to be isothermals. The error resulting from this assumption is usually negligible.

163. Hypothetical Mean Effective Pressure. The hypothetical mean effective pressure is the constant uniform pressure which, if active throughout the piston stroke, would produce the same work area as is bounded by the paths of the hypothetical diagram. There are two cases.

Case I relates to the hypothetical diagram without clearance of Fig. 114. The net area of the diagram equals

$$(344) \quad \text{area } Obcg + \text{area } gcdf - \text{area } aefO.$$

$$(345) \quad \text{Area } Obcg = P_1 V_1,$$

$$(346) \quad \text{area } gcdf = \int_{v_1}^{v_2} PdV = P_1 V_1 \log_e \frac{V_2}{V_1},$$

and

$$(347) \quad \text{area } aefO = P_2 V_2.$$

Let

a = the area of an engine piston in square inches,

A = the area of a rectangle,

h = the altitude of a rectangle,

l = the length of a rectangle, or of an indicator diagram,

$m.e.p.$ = the mean effective pressure,

N = the revolutions of an engine per minute,

P_m = the hypothetical $m.e.p.$ in lbs. per sq. ft.

p_m = the hypothetical $m.e.p.$ in lbs. per sq. in.,

r = the ratio of expansion in an engine cylinder,

V = the volume displaced by the piston in cu. ft. for one stroke (with proper subscript).

¹ See L. S. Marks, *Mechanical Engineers' Handbook*, second edition, pp. 348 and 994, for more complete data on the value of the exponent for steam.

Then the area of any rectangle equals

$$(348) \quad A = hl,$$

or

$$(349) \quad h = \frac{A}{l}.$$

Also

$$(350) \quad r = \frac{V_2}{V_1} \quad (\text{Figs. 114 or 115}).$$

Therefore the mean height or its equivalent, the mean effective pressure which would give a rectangle having the same area and the same length as the hypothetical diagram, would be

$$(351) \quad P_m = \frac{\text{area of the diagram in proper units}}{\text{length of the diagram in proper units}}.$$

The length, fO , of the hypothetical diagram, Fig. 114, may be considered equal to V_2 measured on the axis of volumes. Therefore from (344) to (351) inclusive, the mean effective pressure equals

$$(352) \quad P_m = \frac{P_1 V_1 + P_1 V_1 \log_e r - P_2 V_2}{V_2},$$

or

$$(353) \quad P_m = P_1 \frac{V_1}{V_2} (1 + \log_e r) - P_2.$$

But

$$\frac{V_1}{V_2} = \frac{1}{r}.$$

Therefore

$$(354) \quad P_m = \frac{P_1}{r} (1 + \log_e r) - P_2.$$

Since

$$P = 144 p,$$

equation (354) may be divided by 144, and the result is

$$(355) \quad p_m = \frac{p_1}{r} (1 + \log_e r) - p_2.$$

¹ For multiple-expansion engines the ratio of expansion r for the entire engine equals the total volume of the low-pressure cylinder divided by the volume of the high-pressure cylinder up to cut-off. The clearance volume for the respective cylinders is to be included in each case. The value of the ratio of expansion of any engine is always a number greater than unity.

Case II applies to the hypothetical diagram with clearance and compression as shown in Fig. 115. Since

$$(356) \quad P_m' = \frac{\text{Area } 11_a2456}{V_2 - V_c},$$

it may be shown by steps similar to those carried out for Case I that

$$(357) \quad p_m' = \frac{p_1}{r} (1 + c - cr) + \frac{p_1}{r} (1 + c) \log_e r \\ - p_2 [1 + c (1 - r_c)] - p_2 cr_c \log_e r_c.$$

In (357)

c = proportion of clearance volume to piston displacement,

$$= \frac{V_c}{V_2 - V_c}$$

p_m' = hypothetical mean effective pressure with clearance,

r = ratio of expansion as in Case I,

r_c = ratio of compression = $\frac{V_g}{V_c}$, a number greater than unity.

The mean effective pressure of actual indicator cards is generally obtained by measuring the card area with a *polar planimeter*,¹ Fig. 117, which is an instrument for mechanically integrating areas. It was invented in 1854 by Prof. J. Amsler. Amsler's planimeter was an improvement upon an instrument invented by a Mr. Oppenkoffer in 1827.

The card carrying the area to be integrated is laid flat on a smooth surface and the boundary of the area traced over with the tracing point F , Fig. 117 (a). Meanwhile the pole point P is held stationary. The two planimeter arms FS and CP articulate by a pin joint at C . The record wheel which is shown somewhat enlarged in Fig. 117 (b) is read before and after tracing the boundary, and the difference between the two readings is the

¹ See C. E. Emery, *The Polar Planimeter*, Transactions A.S.M.E., 1885. Vol. 6, p. 496. Report of the British Assn. for the Advancement of Science, 1894, p. 496. J. Y. Wheatley, *The Polar Planimeter*.

area sought. The unit in which the area is measured depends upon the diameter of the record wheel rim, and the effective length of the tracing arm FC . These variables are usually so adjusted that the area is measured in square inches. Adjustable-arm planimeters make it possible to choose one of a number of

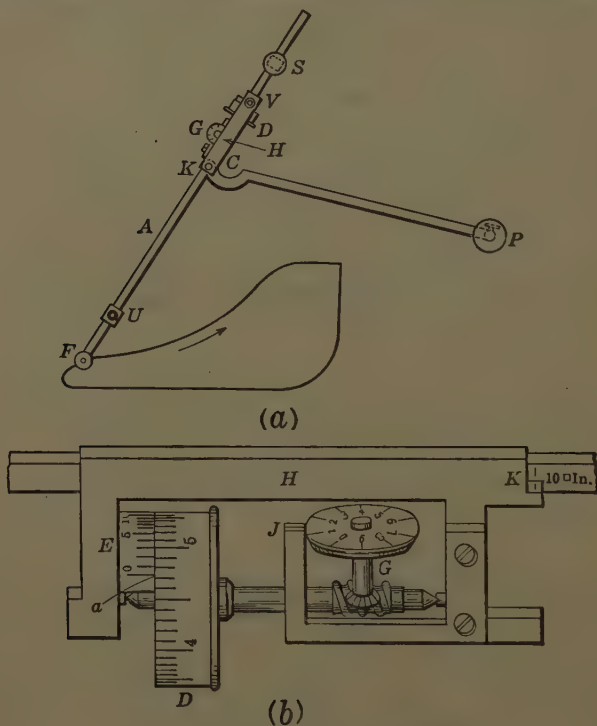


FIG. 117.—POLAR PLANIMETER

possible units. When the area thus obtained in square inches is divided by the length of the area in inches, the mean altitude of the area in inches is obtained. The altitude multiplied by the scale of the indicator spring used in obtaining the indicator diagram gives the mean effective pressure in pounds per square inch.

By equation (65), page 57, work in foot-pounds equals

$$W = PV,$$

and by definition one horsepower equals 33,000 foot-pounds of work per minute. Therefore the indicated horsepower of an

ideal engine giving a hypothetical diagram is

$$(358) \quad \text{I. hp.} = \frac{P_m V}{33\,000}.$$

Now let

a = area of engine piston, sq. ins.,

l = length of piston stroke, ft.,

N = r.p.m.,

$P_m = 144\,p$.

Then

$$(359) \quad V = \frac{alN}{144}.$$

From (358) and (359) the hypothetical indicated horsepower equals

$$(360) \quad \text{I. hp.} = \frac{plaN}{33\,000}.$$

The result given by (360) is for one side of the piston only. For a double-acting engine having the same value of p for both faces of the piston, and neglecting the area of the piston rod on the crank side of the piston, the result would be

$$(361) \quad \text{I. hp.} = \frac{2\,plaN}{33\,000}.$$

The indicated horsepower, as determined by (361), is larger than can be produced by any actual engine because the area of the hypothetical diagram is larger than the area of a diagram from any actual engine.

164. Diagram Factor. The diagram factor, f , is a number less than unity, by which the hypothetical mean effective pressure is multiplied in order to obtain the probable mean effective pressure of an actual engine. In Figs. 118 and 119, the solid-line diagrams represent the same indicator diagram taken from an actual engine operating non-condensing. The difference in altitude of the boiler pressure line and the top of the indicator diagram represents pressure loss due to friction in the steam pipe between the

boiler and engine and the loss through the engine valve due to inadequate port opening. The distance RS , Fig. 118, on the boiler-pressure line represents the length of the admission line

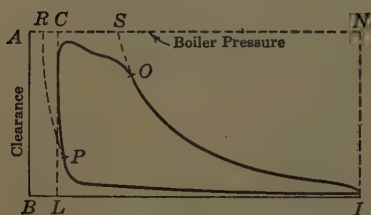


FIG. 118.—ACTUAL INDICATOR DIAGRAM. Net Volume of Steam Admitted equals RS

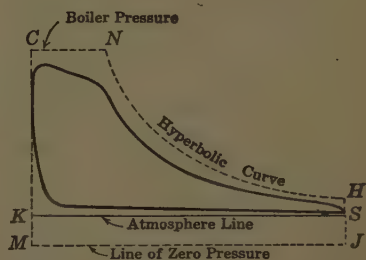


FIG. 119.—ACTUAL AND HYPOTHETICAL DIAGRAMS NON-CONDENSING (From A.S.M.E. Power Test Code)

for the hypothetical diagram. It is obtained by extending the compression and expansion curves of the indicator diagram upward to the boiler pressure as isothermals. The origin for these isothermals is at the intersection of the clearance line AB with

the line of zero pressure which is not shown in Fig. 118. Then CN , Fig. 119, is made equal to RS . Beginning at N the hyperbolic curve NH is constructed with M as the origin. The area $CNHSK$ is the hypothetical diagram without clearance. The ratio of the area of the actual diagram to the area $CNHSK$ is the diagram factor. This statement holds for the non-condensing

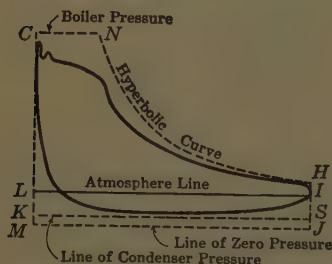


FIG. 120.—ACTUAL AND HYPOTHETICAL DIAGRAM—CONDENSING (From A.S.M.E. Test Code)

diagram of Fig. 119, and for the condensing diagram of Fig. 120 as well. The diagram factor is especially useful to the engine designer, for the probable mean effective pressure of a proposed engine of given type may be readily found by multiplying the hypothetical mean effective pressure by the appropriate factor. Diagram factors range in value from about 0.60 to 0.90, depending

upon the style of engine and valve gear. A few values are given in Table 49 for the hypothetical diagram without clearance.

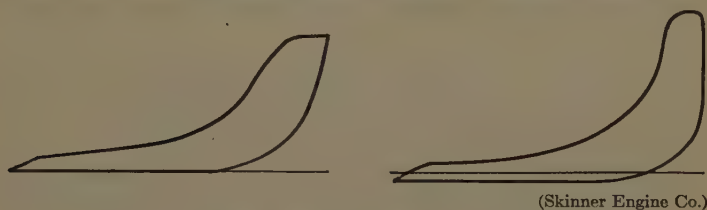


FIG. 121.—CARDS FROM UNIVERSAL UNAFLOW ENGINE

TABLE 49
DIAGRAM FACTORS

TYPE OF ENGINE	<i>f</i>
Small, slide-valve, high-speed engine, non-condensing	0.75 to 0.80
Simple, piston-valve, fly-wheel governor, non-condensing	0.78 to 0.85
Automatic, four-valve, non-releasing gear, non-condensing	0.80 to 0.85
Simple, Corliss-valve, condensing or non-condensing	0.88 to 0.92
Compound, Corliss-valve, condensing or non-condensing	0.78 to 0.83
Poppet-valve, unaflow, condensing engine	0.77 to 0.82
Poppet-valve, unaflow, non-condensing, auxiliary exhaust valves.	0.75 to 0.78

If the factor is for hypothetical diagrams with clearance, Fig. 119, its values are usually between 0.90 and 0.98.¹

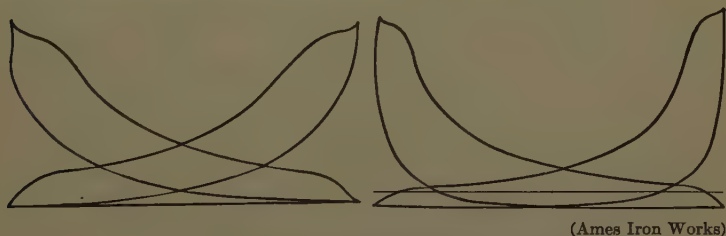


FIG. 122.—CARDS FROM UNAFLOW ENGINE

165. Engine-Cylinder Dimensions. Cylinder dimensions of a steam engine are always specified by stating first the diameter in inches; then the length of stroke in inches is given. Thus a $12\frac{1}{2} \times 16$ engine indicates a simple engine whose cylinder diameter is $12\frac{1}{2}$ inches, and whose length of stroke is 16 inches.

¹ See Marks' and Kent's engineers' handbooks for more complete data on diagram factors.

The numbers $16 \times 28 \times 42$ indicate a compound engine whose high-pressure cylinder is 16 inches in diameter, the low-pressure cylinder is 28 inches in diameter, and the common stroke for both pistons is 42 inches.

The numbers $15 \times 22 \times 48 \times 48$ indicate a triple-expansion engine whose three cylinders are respectively 15, 22, and 48 inches in diameter with a common stroke of 48 inches.

Frequently the size of a steam engine is designated by its horsepower capacity. In that case, it is always the indicated horsepower which is given. But in buying and selling engines, that style is nearly obsolete, since the power of a given engine varies over a wide range with changes in initial and exhaust-steam pressures, the point of cut-off, and engine speed.

The cylinder dimensions of a proposed double-acting engine must be determined from the indicated horsepower formula,

$$(362) \quad \text{I. hp.} = \frac{2 \, p \, l \, a \, N \, f}{33 \, 000},$$

and

$$(363) \quad a = \frac{33 \, 000 \, \text{I. hp.}}{2 \, f \, l \, N \, p}.$$

The piston speed (the distance in feet, which the piston travels in one minute) is a factor much used in engine design. It is usually denoted by the letter s .

$$(364) \quad s = 2 \, l \, N.$$

Hence

$$(365) \quad a = \frac{33 \, 000 \, \text{I. hp.}}{f \, p \, s}.$$

Experience has taught that the greatest economy in steam consumption and engine structure, and the most pleasing operating characteristics, are obtained when piston speeds are fixed between 600 and 1200 feet per minute; 800 to 900 feet per minute being values most frequently selected. It has also been found that the most suitable ratio of length of stroke to piston diameter is from 1.0 to 1.5. With these constants in mind, it is simply a matter of a few trial-and-error solutions of (364) and (365) to obtain suitable cylinder dimensions for a new engine.

166. **Effect of Clearance upon the Ratio of Expansion.** Referring to Figs. 114 and 115 it will be observed that without clearance

$$(366) \quad r = \frac{V_2}{V_1},$$

and for the diagram with clearance

$$(367) \quad r_c = \frac{V_2}{V_1} = \frac{V + V_c}{V_0 + V_c}.$$

Here

r = the ratio of expansion without clearance,

r_c = the ratio of expansion with clearance,

V = total displacement of piston in one full stroke,

V_c = the value of clearance volume,

V_0 = piston displacement up to cut-off.

From these relations it is seen that the value of r must always be larger than r_c in a given cylinder with constant cut-off. The value of r_c becomes smaller and smaller as the clearance volume increases. In general it is desirable to have as large a value for the ratio of expansion as possible in order to obtain high steam economy. Hence for this reason V_c should be made as small as possible. But an engine without clearance is impractical. The piston cannot be permitted to make contact with the cylinder heads at the ends of the stroke because the impacts would soon damage the engine seriously. The steam ports must have some area and length in order to get the steam into and out of the cylinder. It is highly desirable to produce smooth, shockless reversals in the piston motion at the ends of the strokes. Cushioning by compressing a small weight of steam in the clearance space is the most satisfactory method of accomplishing it. For these reasons a certain amount of clearance is desirable and necessary. The amount of clearance which is desirable varies with the size and speed of the engine. It is mechanically impossible to build cylinders with clearance volume less than 1.0 or 1.5 per cent of the piston displacement. Clearance volumes vary from about 1.5 to 15 per cent depending upon the type, size, and speed of the engine. Clearance volumes are ordinarily 8 to 10 per cent

for high speed piston-valve engines and from 4 to 6 per cent in large, moderate, and slow-speed Corliss and poppet-valve engines.¹

167. The Diagram Steam Rate. During each revolution of an engine, a definite weight of steam from the boiler flows in to fill the cylinder volume behind the piston up to the point of cut-off. Then the valve shuts off the steam flow, and the steam caught in the cylinder expands. At release, the cylinder is full of relatively low-pressure steam. The exhaust valve opens and the piston begins its return stroke, sweeping the steam out of the cylinder up to the time of exhaust-valve closure at the beginning of compression. The low-pressure steam remaining in the cylinder is compressed, the pressure rising and the volume decreasing until all this steam is compressed into the clearance space. At this instant, the pressure of the steam in the clearance space has risen well toward that in the boiler. The inlet valve again opens, permitting a fresh supply of steam from the boiler to enter the cylinder. Referring to Fig. 115, the following symbols will be used:

a = area of piston in square inches,

$c = \frac{V_c}{V}$ = ratio of clearance volume to the piston displacement,

l = length of piston stroke in feet,

N = revolutions of engine shaft per minute = r.p.m.,

p = m.e.p. from the indicator diagram,

$r = \frac{V_1 - V_c}{V}$ = ratio of piston displacement up to cut-off to the total piston displacement,

$s = \frac{V_4 - V_c}{V}$ = ratio of piston displacement during compression to the total displacement,

δ_1 = density of steam at cut-off (lbs. per cu. ft.),

δ_2 = density of steam at beginning of compression,

V = piston displacement, cu. ft. per stroke = $(V_2 - V_c)$,

S = steam rate, lbs. of steam per I. hp. per hr.

The pounds of steam in the cylinder at cut-off equals

$$(c + r) V \delta_1.$$

Likewise, the pounds of steam remaining in the cylinder at the be-

¹ See Marks' *M. E. Handbook* for more complete data.

ginning of compression is

$$(c + s)V\delta_2.$$

The net weight of cylinder feed to one end of the cylinder per cycle equals

$$(c + r)V\delta_1 - (c + s)V\delta_2.$$

For the two ends of a double-acting engine and for one revolution the weight of steam supplied the engine is

$$2 V[(c + r)\delta_1 - (c + s)\delta_2].$$

In one hour the total weight is

$$2 \times 60 NV[(c + r)\delta_1 - (c + s)\delta_2].$$

Therefore the steam rate per I.hp. per hour becomes

$$(368) \quad S = \frac{2 \times 60 NV[(c + r)\delta_1 - (c + s)\delta_2]}{\frac{2 plaN}{33\,000}}.$$

This reduces to

$$(369) \quad S = \frac{13\,750}{p} [(c + r)\delta_1 - (c + s)\delta_2].$$

168. The Saturation Line. When hot steam is first admitted from the boiler, the cylinder has just been cleared of the comparatively cold low-pressure steam. The cylinder walls and piston have been cooled appreciably (50° to 200° F.). The temperature is raised by the fresh supply of hot steam and at the expense of some of the heat carried by the new steam. This continual reheating of the cylinder and piston walls causes considerable decrease in the volume of the steam, if it be initially superheated; condensation if it be just dry and saturated, or slightly wet. In the counterflow, slide-valve type of engine, the actual engine steam rate is from 30 to 50 per cent higher than the diagram steam rate computed by equation (369). This large increase is due to *cylinder condensation*. For many years, engineers the world over searched intensively to find some reliable method by which it would be possible to compute the actual steam rate. In 1912 J. P. Clayton, a graduate student of the University of Illinois, found a satisfactory answer.¹ But his solution came too late to be of any great commercial value, since the steam engine has been so largely displaced by steam turbines and Diesel engines.

It is interesting and sometimes useful to determine the changes in the quality of the steam in the cylinder as it expands after cut-off. To do

¹ See Transactions A.S.M.E., vol. 34, p. 75.

so, however, requires a test of the engine in which the actual steam rate and actual indicator diagrams are obtained. The method of determining the saturation line and the quality of the steam from instant to instant during the expansion period can best be developed by use of an illustrative problem. The following data and results were obtained from the test of a simple, double-acting engine operated condensing.

TABLE 50
TEST OF A SIMPLE CONDENSING ENGINE

Cylinder dimensions.....	20 × 30
Full load cut-off occurs at.....	20% of stroke
Release occurs at.....	95% of stroke
Compression during.....	60% of back stroke
Clearance volume.....	4%
Initial pressure, lbs. sq. in. g.....	140
Initial quality.....	0.98
Cut-off pressure, lbs. sq. in. g.....	130
Exhaust pressure, lbs. sq. in. abs.....	2
R.p.m.....	175
I.hp. by indicator card.....	640
Total wt. of steam, lbs. per hr.....	14 000

The actual indicator diagram is reproduced (not to the original scales, however) in Fig. 123 as the area *abcdef*. Two assumptions are made in determining the saturation line *Lrz*: (1) The steam in the cylinder is just dry and saturated (quality = 1.00) at the beginning of compression at *e*. (2) Steam leakage past the valve into the cylinder and past the piston is zero at all times. Then from the test data and from a set of steam tables the following results are computed.

- (1) Lbs. of cylinder feed per stroke = $\frac{14\,000}{2 \times 175 \times 60} = 0.6667$,
- (2) Volume of piston displacement = $\frac{0.7854 \times 20^2 \times 30}{1728} = 5.456$ cu. ft.
- (3) Clearance volume = $5.456 \times 0.04 = 0.218$ cu. ft.
- (4) Total cylinder volume including clearance
= $5.456 + 0.218 = 5.674$ cu. ft.
- (5) Cylinder volume up to cut-off = $5.456 \times 0.2 = 1.091$ cu. ft.
- (6) Total volume up to cut-off = $1.091 + 0.218 = 1.309$ cu. ft.
- (7) Cylinder volume up to release = $5.456 \times 0.95 = 5.183$ cu. ft.
- (8) Total volume up to release = $5.183 + 0.218 = 5.401$ cu. ft.

- (9) Cylinder volume in compression = $5.456 \times 0.6 = 3.274$ cu. ft.
 (10) Total volume in compression
 = $3.274 + 0.218 = 3.492$ cu. ft.
 (11) Density of steam at 2 lbs. per sq. in. abs. = 0.00576 lbs.
 (12) Wt. of steam caught in the clearance
 = $0.00576 \times 3.492 = 0.0201$ lbs.
 (13) Wt. of steam in cylinder during expansion period equals
 cylinder feed plus clearance steam
 = $0.6667 + 0.0201 = 0.6868$ lbs.
 (14) If the steam were dry and saturated throughout the expansion period, its volume at any pressure would be the specific volume (from steam tables) for that pressure multiplied by the total weight of steam in the cylinder during the expansion period as given in Table 51. By

TABLE 51

VOLUMES FOR THE SATURATION CURVE OF FIG. 134

Pressure Lbs. per Sq. In. Abs.	Specific Volume Times Weight	Volume in Cu. Ft.
160	2.830×0.6868	1.946
120	3.725×0.6868	2.559
80	5.470×0.6868	3.757
40	10.497×0.6868	7.205

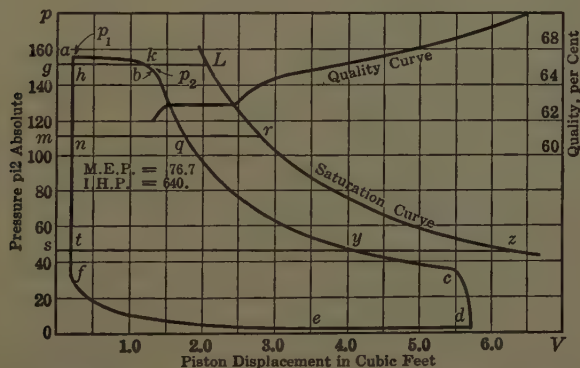


FIG. 123.—SATURATION AND QUALITY CURVES ON THE INDICATOR CARD

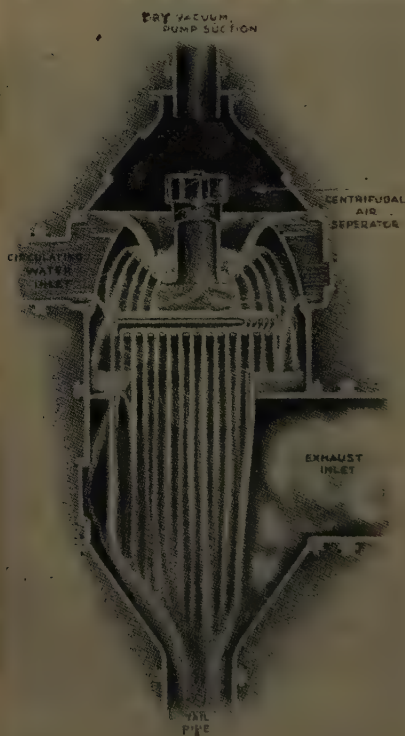
plotting on Fig. 123 these and other similarly computed values of volumes, the saturation curve Lrz may be drawn. At any pressure, g , during the expansion period, the quality

of the expanding steam in the cylinder is equal to the ratio $gk \div gL$. The relatively low value of quality during the expansion period is due almost entirely to cylinder condensation during admission.

169. Condensers. James Watt did not invent the steam engine; he only improved it. But his most important contribution was the invention of the separate condenser in two forms

which are in general use to-day. However, modern condensers are considerably more efficient in transferring heat from steam to water, due to the evolutionary development through which they have since passed.

A sectional view of a modern *jet condenser* is shown in Fig. 124. In this style of condenser, the exhaust steam from an engine is brought into direct contact with a multiplicity of small jets of cold water. The mixture of condensed steam and cooling water is then carried away through a tail pipe. If the tail pipe extends vertically downward for about 34 feet, as shown in Fig. 125, a



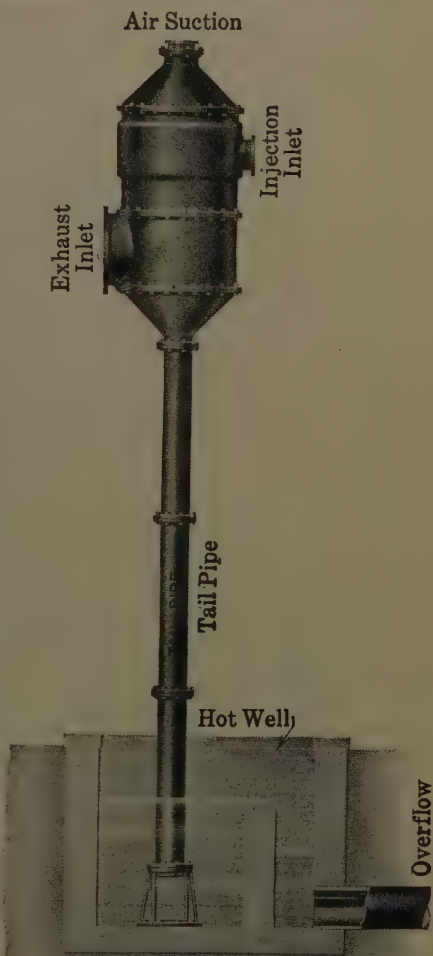
(Worthington Pump and Machinery Corporation)

FIG. 124.—JET CONDENSER

vacuum of 26 to 28 inches of mercury is automatically produced in the condenser by the siphoning action of the water in the tail pipe, which also removes the non-condensable gases. This form of condenser is known as the *barometric condenser* and was so named by Watt. The form of condenser shown in Fig. 126, in

which the discharge suction in the tail pipe is produced by a pump, is known as the *low-level jet condenser*.

The greater part of the vacuum in any condenser is produced by the large reduction in volume when the water in the vapor phase changes to the liquid phase. One pound of dry and saturated steam at a pressure of one pound per square inch absolute occupies 333 cubic feet. At the same temperature and pressure, the volume of one pound of water is 0.0161 cubic feet. Thus the volume of the vapor is 2000 times the volume of the liquid. This large volume change which accompanies condensation of steam in a condenser produces a very respectable vacuum. Boiler feed water generally contains small quantities of dissolved oxygen and other gases. Air leakage into a condenser and the connected piping cannot be entirely eliminated. Therefore, a relatively small vacuum pump¹ of some kind is required to maintain the vacuum in any style of condenser.



(Worthington Pump and Machinery Corporation)
FIG. 125.—BAROMETRIC CONDENSER

¹ Limitations of space prevent more than the mention here of pumping machinery. For information on pumps see A. M. Greene, *Pumping Machinery*, R. L. Daugherty, *Centrifugal Pumps*, N. Swindon, *Modern Theory and Practice of Pumping*.

In *surface condensers*, illustrated in Figs. 127 to 130 inclusive, the cooling water and the steam are in separate compartments. The cooling or circulating water flows through a large number of thin brass or bronze tubes which are housed in a cast-iron or steel shell. As shown in Fig. 127, the circulating pump at the right forces the cooling water into the right end of the condenser head. The water is guided into the lower half of the bank of



(Worthington Pump and Machinery Corporation)

FIG. 126.—JET CONDENSER AND PUMP

condenser tubes by a baffle plate between the head and the tube plate. The water returns to the right-hand and again through the upper half of the group of tubes, and is discharged to the sewer or discharge canal. The exhaust inlet is located in the top of the shell midway between the two ends. The steam passes over the outside of and among the condenser tubes. The condensate gathers at the left end of the condenser, and is removed by the wet vacuum pump as shown. In very large condensers, as shown

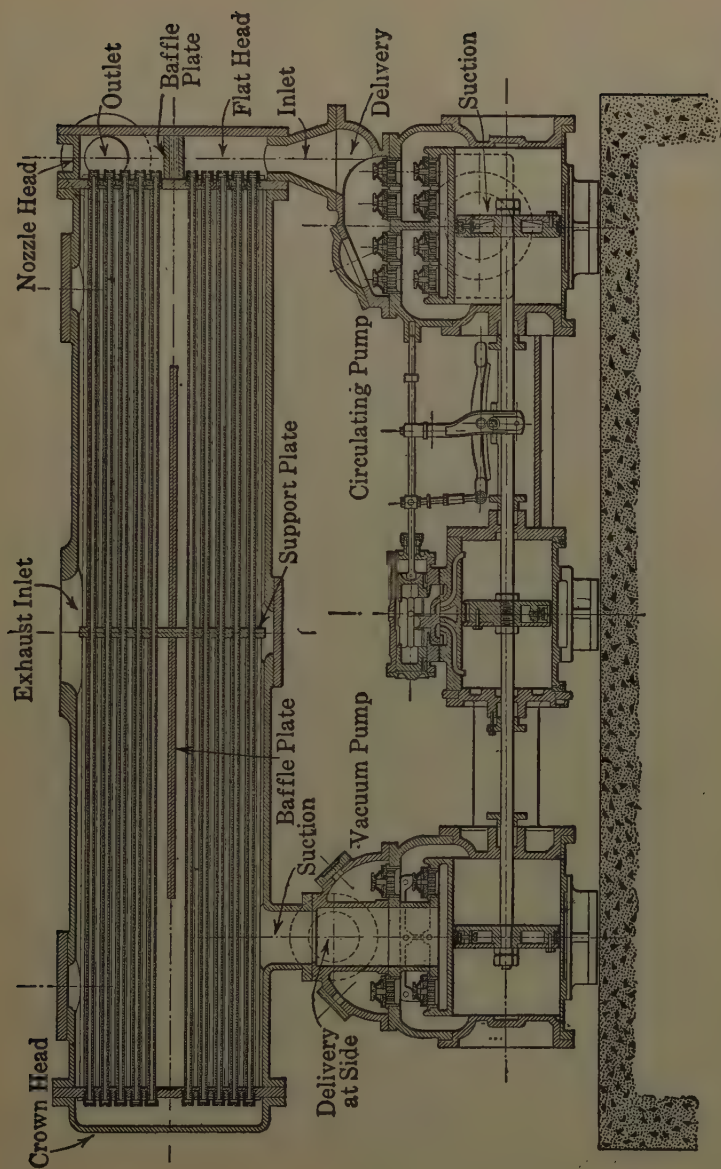
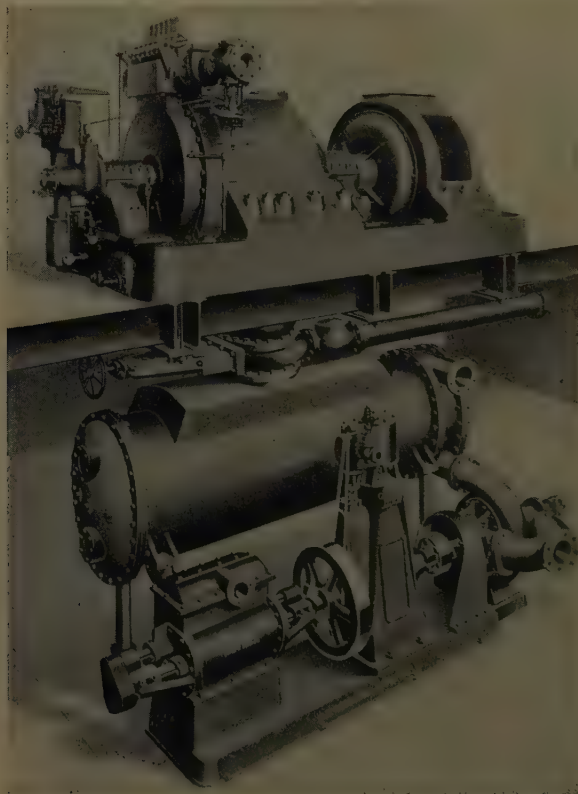


FIG. 127.—SURFACE CONDENSER MOUNTED ON COMBINED AIR AND CIRCULATING PUMP
(Worthington Pump and Machinery Corporation)

in Figs. 129 and 130, the condensate and the air are generally removed by separate pumps. So far as possible, the contraflow principle is used in arranging the direction of flow of water and steam. By this means the rate of heat transfer can be maintained at a high value, and at the same time the condensate can be



(C. H. Wheeler Mfg. Co.)

FIG. 128.—STEAM TURBINE AND SURFACE CONDENSER

cooled to within about 8° F. or 15° F. of the temperature of the cooling water as it enters the condenser. But the more important result in high-vacuum condensers is that the highest possible vacuums are attained; since in the presence of water vapor no vacuum can be higher than that corresponding to the temperature of the boiling point of the condensed steam.

The chief advantage of using a condenser is due to the considerable reduction of the exhaust pressure against which the piston expels the exhaust steam. This reduction of pressure adds

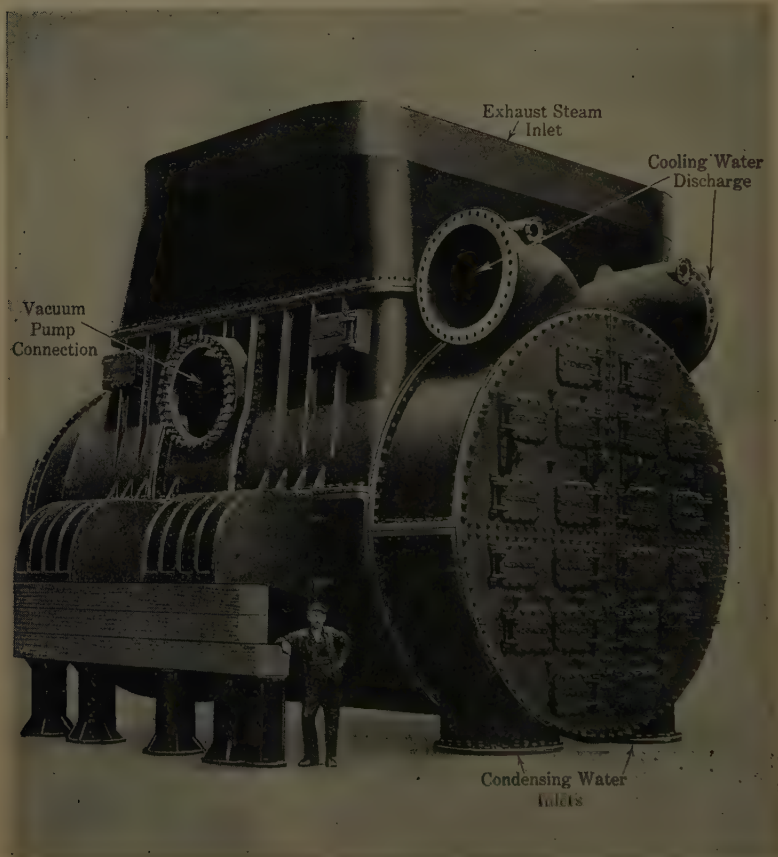


(C. H. Wheeler Mfg. Co.)

FIG. 129.—TWO-PASS SURFACE CONDENSER (50 000 Square Feet)

from 10 to 25 per cent to the effective work area of the indicator card as may be seen by inspecting the diagrams shown in Figs. 120, 121, and 122. This addition to the work area is obtained by

the expenditure of a comparatively small amount of power to operate the vacuum pump and the water-circulating pump. However, the gain with small engines is not sufficient to pay for the additional equipment and maintenance costs. By long ex-



(Worthington Pump and Machinery Corporation)

FIG. 130.—EXTERIOR VIEW OF 50 000-SQUARE-FOOT CONDENSER

perience, it has been found that there is little or no economic gain in operating condensing engines developing less than 200 or 300 horsepower, depending upon local conditions at each plant.

Fig. 128 illustrates the general arrangement of a small turbine-generator unit and its condenser. The general plan of placing the condenser as close as possible to the engine or turbine is considered

the best practice, because there is less pressure loss, and less opportunity for air leakage, thus making possible, under favorable circumstances, vacuums within 0.5 inch of mercury of the barometer reading. The very large condensers illustrated in Figs. 129 and 130 are used for steam turbines of 35,000 to 75,000 kilowatts capacity. For engine and turbine units from 500 to 10,000 horsepower, the amount of required condensing surface available in the tubes varies from about 1.5 to 2.5 square feet for each rated horsepower of the unit. For large turbine units the required condenser surface varies from 0.70 to 2.00 square feet, with the average being about 1.0 square foot per kilowatt of rated capacity.¹ A condenser surface of 0.65 square foot per kilowatt of rated capacity has been found adequate in a few cases for the largest modern turbine-generator units from which steam is extracted at three or four points for feed water heating (see pages 446-450 herein).

In all modern central-station plants, *surface condensers* are used for two reasons. Higher vacuums may be attained, and what is more important, the condensate is pure distilled water which can be returned direct to the boiler with nearly all of its heat of the liquid. The condensate from turbines contains no oil or scale-forming compounds.

Jet condensers are generally employed in small plants (below 5000 horsepower), and with steam engines more frequently than with turbines because the condensate from engines contains cylinder oil which can be completely filtered out only by an expenditure of considerable money for special equipment.

170. Entropy Continued.² For the program of constant pressure carried out in steam generation in a boiler, equation (211), page 121, becomes

$$(370) \quad N_2 - N_1 = c_p M \log_e \frac{T_2}{T_1}.$$

¹ For more complete information on the theory, design, and performance of condensers see G. F. Gebhardt, *Steam Power Plant Engineering*; and R. J. Kaula and I. V. Robinson, *Condensing Plant*.

Ejectors Monopolize Vacuum Field, Power Plant Engineering, Aug. 1, 1929, p. 853.

² Before proceeding, it would be advantageous for the student to turn back and review § 85, p. 120.

When a substance absorbs or gives out heat energy, but without change of state, temperature changes, only, occur. In that case equation (370) may be written thus.

$$(371) \quad N = \pm B \log_e \frac{T_2}{T_1}$$

in which N is understood to represent changes in entropy ($N_2 - N_1$) corresponding to the changes in temperature as indicated,

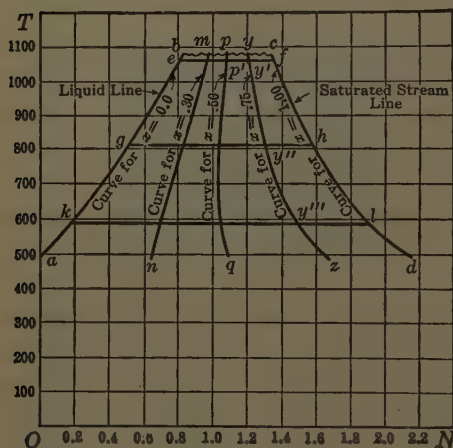


FIG. 131.—CONSTANT-QUALITY CURVES ON THE TN -PLANE

similar to the line ab . When the sign is negative, the path is similar to curve cd . Lines ab and cd are logarithmic curves.

During a change of state upon heat addition, there is no temperature rise, and the entropy change varies with the weight of the substance whose state is changed. This change is represented in Fig. 131 by straight horizontal lines as gh whose equation may be written in the form

$$(372) \quad N = \frac{Mrx}{T}$$

in which

N = the change in entropy,

r = the latent heat required to completely change the state of unit weight of the substance,

and B is a constant involving the weight and the mean specific heat factors. When the second member of (371) has the positive sign; that is, when the path followed on the TN -plane (Fig. 131) is such that the absolute temperature of the substance rises as heat energy is added to it, or the temperature falls as heat is given up, the general direction of the TN -curve will be

M = weight of the substance which changes state,

T = Absolute temperature at which the change of state occurs,

x = The decimal part of M which has been changed in state at any instant.

The number r may represent the latent heat of fusion, or latent heat of evaporation. For any given substance and pressure, the numbers M , r , and T are constant.

The number x corresponds to the term, *quality of steam*, as used in a narrower sense. In the broadest sense, x represented at any instant that decimal fraction of a given weight of a substance which has been changed to the new state. The value of x increases progressively from zero to one in direct proportion to the heat flow when the flow occurs at constant temperature as it always does during any change of state. If heat is absorbed by the substance, x is given the positive sign; if taken away the negative sign. Therefore (372) becomes

$$(373) \quad N = (\pm x) B' = \pm B'x$$

in which

$$(374) \quad B' = \frac{Mr}{T}, \quad \text{a constant.}$$

Therefore

$$(375) \quad N \sim x.^1$$

In Fig. 131, the curve ab represents the liquid line for water. (In dealing with TN -figures for water and steam M is usually taken equal to one pound). The length of the line ef represents to scale the change in entropy when one pound of water at its boiling point is converted into dry and saturated steam by the addition of the latent heat of evaporation at the temperature indicated. Likewise, the lines gh and kl represent to scale the increase in entropy when one pound of water is evaporated from and at its boiling temperature as indicated. These horizontal lines represent the history of the temperature-entropy change as evaporation proceeds. The curve chl , drawn through the ex-

¹ ($\sim$) is used as the sign of variation. Equation (375) is read " N varies as x ."

tremities of the lines ef , gh , and kl , is known as the saturation line. If at any point as y' , on ef or y'' or gh , the addition of heat stops, only a portion of the latent heat of evaporation has been added and a mixture of liquid and vapor constitutes the weight of the H_2O in the ratio of

$$(376) \quad \frac{\text{fractional part of } H_2O \text{ in vapor}}{\text{one pound of the mixture}} = \frac{ey'}{ef} = \frac{gy''}{gh}.$$

The ratio stated in (376) gives the varying values of the quality of the steam as evaporation begins and proceeds to the point of dry and saturated steam along the line chd . Curves so drawn within the steam dome on the TN -plane as to maintain a constant ratio

$$(377) \quad \frac{ey'}{ef} = \frac{gy''}{gh} = \frac{ky'''}{kl}$$

are called *constant-quality lines*, and the curve $yy''z$ represents the relation between temperature and entropy of a mixture of steam and water having a constant quality.

Constant-volume curves may be drawn, within the steam dome on the TN -plane, in a manner somewhat similar to the constant-quality curves.

Let

v_s = volume of one lb. of dry and saturated steam at a given pressure in cu. ft.,

v = volume of one lb. of a mixture of water and its vapor at the same pressure,

σ = volume of one lb. of water at the boiling point. (The value of σ varies from 0.016 to .05 depending upon the pressure),

x = quality of the mixture.

Then

$$(378) \quad x = \frac{\text{increase in the volume of the mixture}}{\text{increase in the volume of 1 lb. of dry steam}},$$

$$(379) \quad x = \frac{v - (1 - x)\sigma}{v_s - \sigma} = \frac{v - \sigma}{v_s - \sigma}.$$

because the product $x\sigma$ may be neglected. By so controlling conditions as to hold v constant and substituting values of v_s corresponding to different absolute temperatures, the manner in which x varies with temperature during a constant-volume change may be computed. The constant-volume curve may then be plotted on the TN -plane by employing the values of x computed by substituting (379) in (373). Referring to Fig. 132, it is seen that

$$(380) \quad x_1 = \frac{bl}{bc} = \frac{v - 0.017}{v_s'},$$

$$(381) \quad x_2 = \frac{em}{ef} = \frac{v - 0.0165}{v_s''},$$

$$(382) \quad x_3 = \frac{gn}{gh} = \frac{v - 0.016}{v_s''' }.$$

By connecting the points l , m , and n , set off in accordance with the above computed values, the curve of constant-volume change (isometric) is located on the TN -plane.

In a similar manner, the curve of constant enthalpy may be drawn. For wet steam it will be recalled that the enthalpy is

$$(383) \quad H_w = xr + h \quad (\text{see symbols of p. 116}).$$

By making the value of H constant, a succession of values of the quality may be computed as shown in Fig. 132.

$$(384) \quad x_1 = \frac{bl}{bc} = \frac{H_w - h_1}{r_1},$$

$$(385) \quad x_2 = \frac{ep}{ef} = \frac{H_w - h_2}{r_2},$$

and

$$(386) \quad x_3 = \frac{gh}{gj} = \frac{H_w - h_3}{r_3}.$$

These equations give the variations of x with pressure, or they give the corresponding values of absolute temperature, and the curve of constant total heat on the TN -plane, as $lphq$, is located.

For wet steam, the temperature at any pressure is constant during evaporation. Hence, on the TN -plane, constant-temperature and constant-pressure changes *within the steam dome* are represented by straight horizontal lines as bc and ef , Fig. 132. *Within and without the steam dome*, constant entropy or adiabatic curves are straight vertical lines.

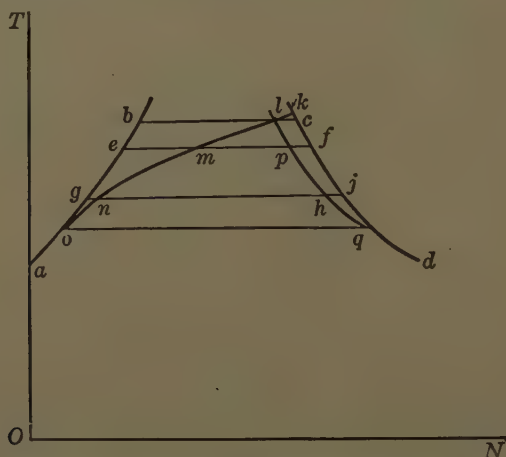


FIG. 132.—CONSTANT-VOLUME AND CONSTANT-ENTHALPY CURVES ON THE TN -PLANE

The equations and the forms of six curves on the TN -plane for vapors have now been developed. These are the six special curves obtained when heat and work changes occur while some one of the six cardinal properties remains constant. The nomenclature which has been applied to the PV -curves may be applied to these TN -curves with equal appropriateness: namely, *isothermal*, *adiabatic*, *isometric*, *isobaric*, etc.

171. Transformation of PV -Diagrams into Equivalent TN -Diagrams for Steam. It is assumed for the purpose of the transformation that the engine cylinder has no valves, and that it contains throughout the cycle one pound of H_2O of which varying proportions are in the vapor state, the remainder of the pound of substance being in the liquid state whose volume is negligible. Hence the quality of the mixture of steam and liquid is continu-

ally varying throughout the cycle, but its weight is constant at one pound. During the expansion period the values of quality are real, that is, they are found to exist in the actual cylinder. During the remainder of the cycle, they are imaginary. Thus

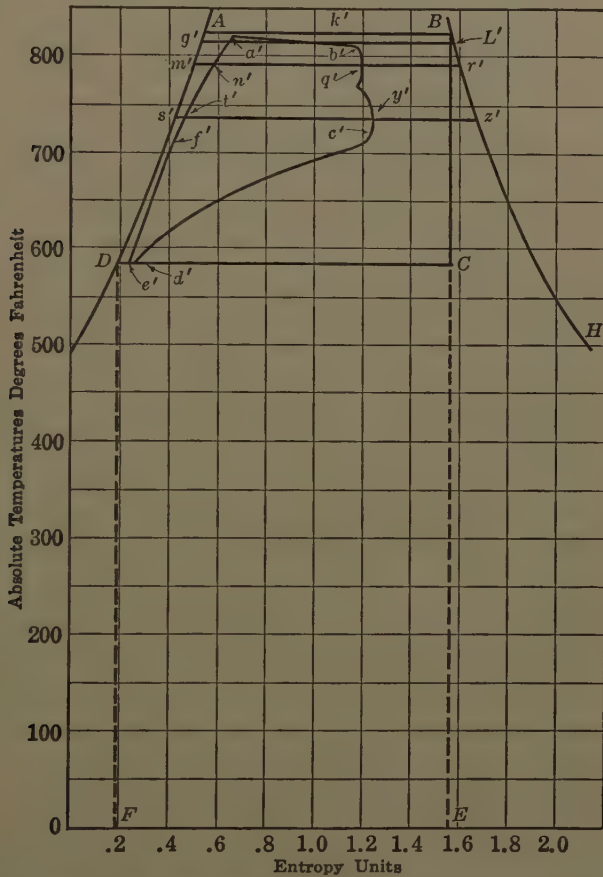


FIG. 133.—INDICATOR CARD ON THE TN -PLANE (See Fig. 123 for Indicator Card)

referring to Fig. 123, the ratio $mq \div mr$, measured off to scale as linear distances, represents the quality of the steam in the cylinder during the expansion period at the instant when the pressure is 110 pounds per square inch absolute. This quality is a real number. Likewise the ratio $mn \div mr$ represents the value

of the quality in the cylinder at the instant during the admission period when the pressure is 110 pounds. This quality is imaginary in that it does not exist in the actual engine. But for the purposes of this analysis it is immaterial whether, during parts of the cycle, steam is expelled from the cylinder and then taken in again, or whether a fixed weight of it remains throughout the cycle, at certain instants nearly all in the vapor state, at other instants nearly all in the liquid state. Thus by employing the values of quality, real or imaginary, throughout the cycle an equivalent *TN*-diagram may be constructed point by point.

In Fig. 133 the ordinate scale is absolute Fahrenheit temperatures, and the abscissa scale is entropy units for one pound of H_2O liquid or vapor, taken from the steam tables. The curve *DA* is the liquid line, and the curve *BH* is the saturation line. The lower-case letters with prime marks indicate points in the *TN*-diagram corresponding to the same points in the *PV*-diagram, Fig. 123. The liquid line *DA*, Fig. 133, is the base from which quality values are measured the same as the axis of ordinates, *op* (Fig. 123), is the base from which quality values on the *PV*-plane are determined. Thus the ratios $m'q' \div m'r'$ and $m'n' \div m'r'$, of Fig. 133, are laid off equal to the corresponding ratios $mq \div mr$ and $mn \div mr$ of Fig. 123. On the *PV*-plane these quantities are measured in volume units, on the *TN*-plane they are measured in entropy units. Thus throughout the cycle, at each steam pressure or the equivalent temperature, there are two

TABLE 52
DATA AND RESULTS FOR *TN*-DIAGRAM

(1) <i>p</i>	(2) <i>t</i>	(3) <i>T</i>	(4) <i>n_g</i>	(5) <i>n_c</i>	(6) <i>x_c</i>	(7) <i>x_gn_g</i>	(8) <i>n_l</i>	(9) $\frac{n_l}{+ n_c x_c}$	(10) <i>x_c</i>	(11) <i>x_cn_g</i>	(12) $\frac{n_l}{+ x_c n_g}$
160	364	824	1.56				.521				
154.7	361	821		1.050	.108	.114	.517	.631	.108	.114	.631
150	358	817	1.57	1.055	.552	.582	.514	1.096	.1052	.111	.625
145	356	815	1.572	1.061	.614	.652	.511	1.163	.1014	.1076	.619
140	353	813	1.575	1.067	.627	.669	.507	1.176	.098	.1048	.612
120	341	801	1.587	1.095	.629	.689	.492	1.181	.085	.093	.585
100	328	787	1.602	1.128	.628	.708	.474	1.182	.071	.081	.555
60	293	752	1.643	1.216	.650	.790	.427	1.217	.0447	.054	.481
40	267	730	1.676	1.284	.659	.846	.392	1.238	.030	.039	.431
35	259	719		1.306	.660	.862	.381	1.243	.027	.035	.416
30	250	710		1.331	.593	.789	.368	1.157	.024	.032	.400
20	228	688	1.732	1.397	.412	.575	.336	0.911	.026	.037	.373
10	193	653		1.504	.216	.324	.283	0.607	.027	.041	.324
2	126	586	1.919	1.743	.048	.083	.175	0.258	.029	.051	.226

values of quality. By transferring a sufficient number of these quality points over to the TN -plane (usually about 12 or 15 points are sufficient) the equivalent TN -diagram may be constructed. A suggested form of arranging the data and computation is shown in Table 52.

In Table 52, the symbols at the head of the columns represent the following quantities:

- p = the steam pressure in lbs. per sq. in. abs.,
- t = boiling point temperature, degrees F., corresponding to the pressure as read from steam tables,
- T = absolute steam temperature, degrees F., corresponding to t ,
- n_s = total entropy of 1 lb. of dry and saturated steam read from steam tables, corresponding to pressure p ,
- n_e = entropy of evaporation, as read from steam tables for dry and saturated steam, corresponding to pressure p ,
- n_l = entropy of the liquid, read from steam tables,
- x_e = quality of the steam as computed from Fig. 123 between a and d along the path $aqycd$,
- x_c = quality of the steam along the path $defna$.

The area $a'b'q'y'c'd'f'a'$ (Fig. 133), represents correctly to scale the same energy in B.t.u. as is represented in foot-pounds by the area of the indicator diagram of Fig. 123. Likewise, the area $ABCD$ represents the B.t.u. of work done by one complete cycle in the ideal Rankine engine. The discrepancies between the Rankine engine and actual engine TN -areas represents the location and magnitudes in B.t.u. of the energy losses in the actual steam engine.

In Fig. 134, the Carnot ideal cycle for dry and saturated steam is shown on both the PV -plane and the TN -plane. Fig. 135 shows the Rankine ideal cycle on the two planes. Fig. 136 shows the Rankine cycle with release occurring before expansion reaches the exhaust-pressure line.

The ideal Carnot and Rankine engines operate on a closed cycle and a closed cylinder having no valves; hence the complete cycle takes place entirely within the engine cylinder with a

constant weight of the substance, in fact the same molecules, and all the molecules in the cylinder at the beginning remain in the cylinder throughout the series of changes which constitute the cycle. In the actual engine, the pressure, volume, and heat changes are those for a varying weight of steam. During the expansion period bc , Fig. 123, the weight of steam in the cylinder remains constant. During the compression period ef , the weight is likewise constant but different. During the period fab , the weight of steam increases while it decreases along cde . The

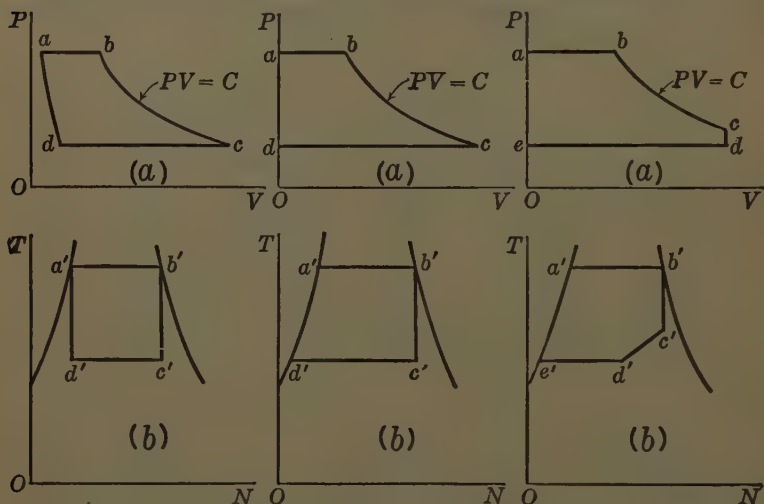


FIG. 134

FIG. 135

FIG. 136

CARNOT AND RANKINE CYCLES

quality of the steam, during the periods bc and ef , may be determined by comparing the volumes in the cylinder with the volumes of the same weight of dry and saturated steam as found in steam tables. But even approximately accurate information as to the true values of quality during the admission and exhaust periods cannot be determined. The qualities used for these periods in transferring the PV -diagram to the TN -plane are purely fictitious or imaginary values. However, the areas under these paths represent foot-pounds of work. It is therefore possible to construct a TN -diagram whose areas represent the corresponding quantities of heat energy. Such a diagram will not give the thermal history

of any definite quantity of steam. It is merely the projection of the PV -diagram to the TN -plane. But the areas of the new figure truly represent, to scale, the same quantities of energy in B.t.u. units as are involved in the corresponding PV -areas where the energy quantities are expressed in foot-pound units. And the differences between areas of the ideal and actual diagrams on the TN -plane indicate in a most impressive manner the exact portions of the cycle in which the various losses occur. This constitutes the chief value of the TN -diagram in studying engine cycles.

172. Mollier and Ellenwood Diagrams. In 1904, Dr. R. Mollier published ¹ a new diagram which has become known universally as the *Mollier Diagram* (Fig. 137). This diagram has proved itself to be a great convenience and a time-saver in solving many problems involving the properties of vapors, especially the vapor of water. Specific values of total heat above 32° F. and total entropy of the vapor are plotted as coördinates. Two sets of intersecting curves, one for pressure and the other for quality or superheat are then drawn. Thus the values of four of the six cardinal properties appear upon the diagram. A point is located on this plane when any two of the four properties are known, then the values of the other two properties corresponding to the location of the point may be determined immediately by inspection. Constant-volume curves could be plotted on this same plane but their intersection with some of the other curves make such very acute angles that the exact values of the properties cannot be determined with a satisfactory degree of precision. To meet this objection, Dr. Mollier prepared a second diagram having specific values of total heat and absolute pressure as coördinates. Constant volume and constant quality curves are then drawn which give suitable intersections. Diagrams having sufficiently large scales to give readings of commercial accuracy are usually supplied with steam tables. Both are in reality Mollier diagrams. But the total heat-entropy diagram is of much greater usefulness. It is a great time saver in solving problems involving constant heat changes (throttling calorimeter

¹ See Zeitschrift des Vereines deutscher Ingenieure, 1904, vol. 48, p. 271.

problems), and adiabatic changes. Fig. 137, drawn to a reduced scale, illustrates its possibilities. Other Mollier diagrams for other vapors, among them being ammonia, carbon dioxide, sulphur dioxide, and mercury,¹ are available.

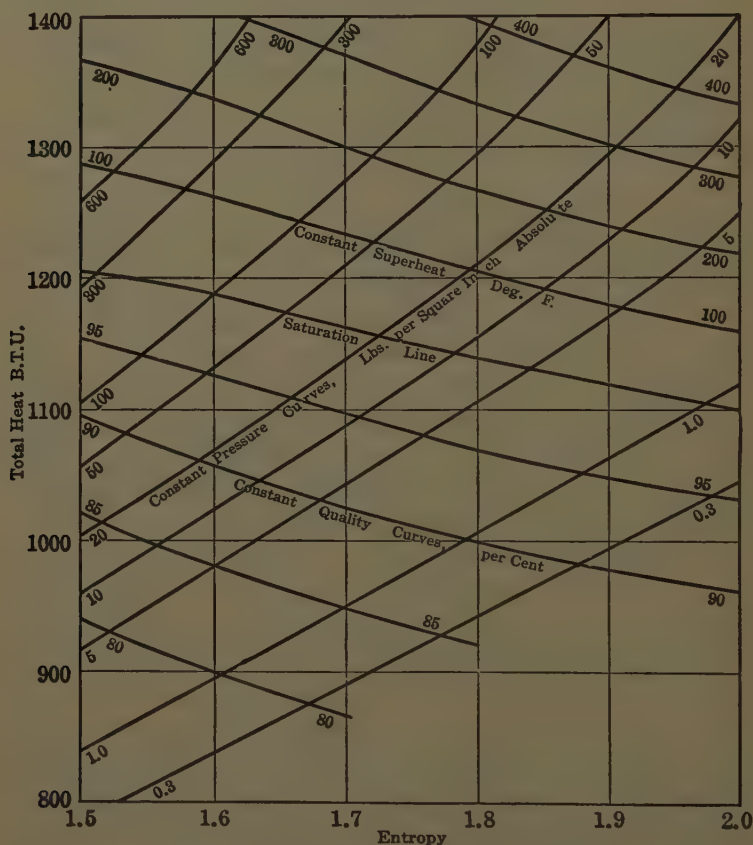


FIG. 137.—MOLLIER DIAGRAM FOR STEAM

In 1914, Professor F. O. Ellenwood² published a diagram for steam which is especially useful in solving steam turbine problems. On the Ellenwood diagram, the ordinates are values of total heat,

¹ See *Miscellaneous Publications No. 52*, 1923, and *Circular No. 142*, 1923, of the U. S. Bureau of Standards for ammonia. Proceedings, Institution of Mechanical Engineers, Oct., 1914, for carbon dioxide and sulphur dioxide. Transactions, A.S.M.E., vol. 46, 1924, p. 273, for a mercury Mollier diagram.

² John Wiley and Sons, Pub.

and the abscissas are specific volumes. Isentropic, isobaric, isoquality, and isosuperheat curves are drawn upon the chart as aids in solving problems involving heat changes. The Ellenwood diagram is especially useful in determining the necessary cross-sectional areas through the various steam passages in turbine design.

173. Methods of Governing Steam-Engine Speeds. The function of a governor is to automatically control the rotative speed of an engine shaft within certain desirable or necessary limits in accordance with the varying load demands. There are two kinds of speed fluctuations to be controlled. Since the force applied to the piston during a single stroke varies from instant to instant due to the expansive use of the steam, there may be rather large variations of speed during a single revolution. This kind of variation is held within desirable limits by the flywheel by virtue of whose mass, advantageously disposed, energy is stored during those instants in a stroke when the force driving the piston is above the average, and the energy is then given back during the portion of the stroke when the driving force behind the piston falls below the average value. By controlling the size and weight of the flywheel and reciprocating parts of the engine, momentary speed variations may be held within the necessary limits.

To a limited extent, the flywheel functions as a governor. Then there is the tendency for the engine speed to vary through very wide limits when the load demand changes. It is the office of the governor to supply varying amounts of steam to meet the changing load conditions, and to keep the engine speed within reasonable limits. In driving steel-mill rolls, speed variations of

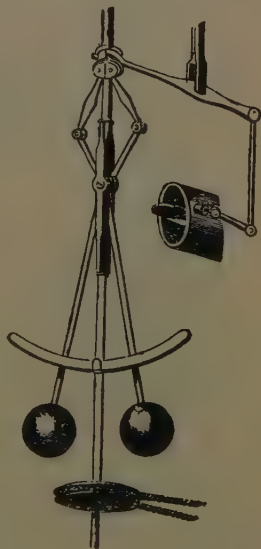


FIG. 138.—WATT'S THROTTLING GOVERNOR

5 or 10 per cent above or below normal from no load to full load may be permissible. In synchronous operation of engine-driven alternating-current generators, and in driving spindles in cotton or woolen mills, speed variations of even one per cent may be undesirable.

Two types of governors are employed: *throttling governors* and *automatic cut-off governors*. The flyball throttling governor



(The Pickering Governor Co.)

FIG. 139.—BALANCED - VALVE THROTTLING GOVERNOR

restricts the steam-pipe area to the passage of steam from instant to instant by a gate or valve. It is generally actuated by a pair of heavy pendulum-like weights, rotating about a shaft. One type of flyball governor is shown in Fig. 88. The changing centrifugal force due to increasing speed causes the balls to describe a larger circle. This movement is communicated through a suitable linkage to the throttling valve located in the governor body. Thus the steam supply is reduced in proportion to the duty required at the moment, and the engine speed is maintained sufficiently uniform. A modern form of the flyball throttling-governor is shown in Figs. 139 and 140.

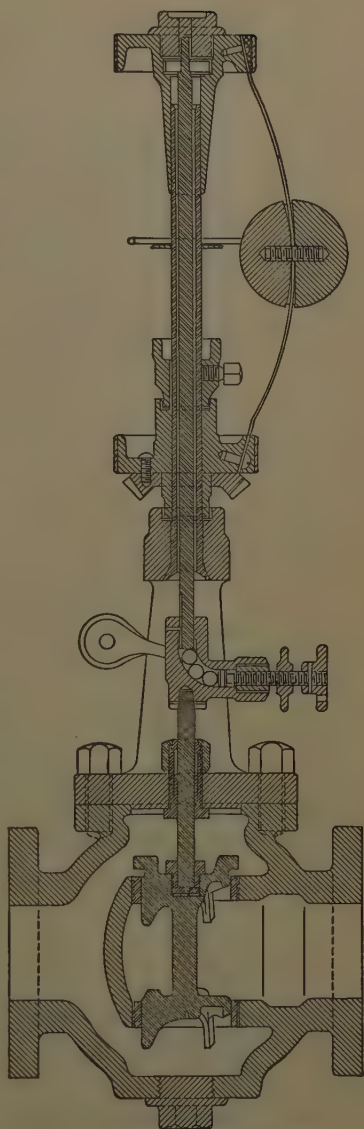
The throttling governor was one of Watt's inventions (Fig. 138), and was first used by him about 1785.

The effect upon the indicator-diagram of changing the initial pressure of the steam to meet the changing load demand is shown in Fig. 141. The admission and expansion line *abc* represents the conditions for maximum engine load. As the load falls, the governor throttles the steam supply, reducing the pressure to the engine which gives the reduced card area under the lines *a'b'c'* or *a''b''c''* or any variation in between as the load conditions

require. With this type of governor, the point of cut-off remains fixed for all loads.

The modern type of governor, the automatic cut-off, permits steam at full pressure to enter the cylinder at all times. But the point of cut-off is varied to give the necessary card area to meet the changing load. The operating mechanism is usually placed in the flywheel. One form of automatic cut-off governor is shown in Fig. 84, p. 290. Weights and linkages are so arranged as to vary the stroke of the valve stem, thus opening the steam ports a longer or shorter period. The effects of this type of governor upon the indicator card is shown in Fig. 142. The initial steam pressure remains fixed or nearly so along the line *ab*. Then the point at which steam admission is cut off varies as shown to meet the changing load conditions.

One of the earliest forms of automatic cut-off valve gears was developed by Zachariah Allen in 1834. In 1842 F. E. Sickels was granted a United States patent on a form of independent cut-off gear for walking-beam engines with a dash-pot arranged to cushion the seating of the closing valve. George H. Corliss¹ at-



(The Pickering Governor Co.)

FIG. 140.—BALANCED-VALVE THROTTLING GOVERNOR

¹ R. H. Thurston, *A Manual of the Steam Engine*, p. 88.

tached the governor to a releasing latch on his rocking-valve gear, which permitted full port opening up to cut-off with very rapid closure of the port. This valve was brought gently to rest by use of the Sickels dash-pot. One form of the Corliss gear and dash-pot is shown in Figs. 87 and 88. The modern forms of the high-speed automatic cut-off engine was first brought out by the partnership firm of Charles T. Porter and John F. Allen ¹ about 1868. However, high-speed engines did not come into general use until after 1890. The first arrangement of an automatic cut-off governor, mounted within the flywheel on the main shaft, was used by J. W. Thompson about 1875 in what became known as the Buck-eye engine.

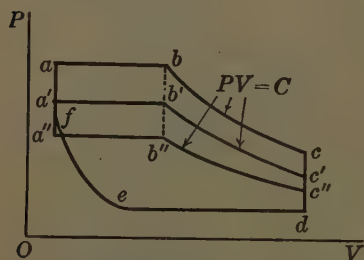


FIG. 141.—INDICATOR DIAGRAM FOR THROTTLING GOVERNOR

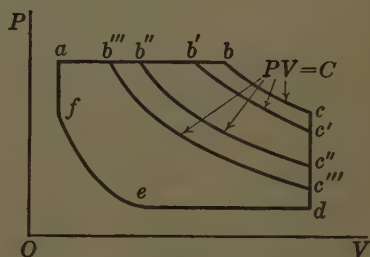


FIG. 142.—INDICATOR DIAGRAM FOR VARIABLE CUT-OFF GOVERNOR

The advantages of each method of governing may be compared as follows:

THROTTLING

- (1) Low first cost,
- (2) Simplicity of mechanism,
- (3) More uniform crank effort.

CUT-OFF

- (1) Closer speed regulation,
- (2) Larger ratio of expansion,
- (3) Lower terminal pressure.

174. Performance Results of Steam Engines. There are a number of ways of measuring and comparing steam-engine performances:

- (1) by means of the indicated and developed horsepower (I. hp. and D. hp.),

¹ Charles T. Porter, *Engineering Reminiscences*.

- (2) by the horsepower consumed in internal friction (F. hp.),
- (3) by the steam rate, *i.e.* the pounds of steam supplied to the engine to produce one horsepower hour (I. hp. or D. hp.),
- (4) by the heat rate, *i.e.* the B.t.u. supplied in the steam to produce one horsepower hour (I. hp. or D. hp.),
- (5) by means of six ratios called efficiencies.

The *indicated horsepower* is computed from indicator cards taken from the engine as already described (page 331). The *developed horsepower* is the horsepower delivered by the crank shaft of the engine for use outside the engine itself. The difference between the indicated horsepower and the developed horsepower is the *friction horsepower*, which represents the energy dissipated by friction in the various bearings and moving parts of the engine.

The developed horsepower may be measured by applying a brake or an absorption dynamometer to the flywheel or to the engine shaft. If the engine is used to drive an electrical generator the generator output and losses may be readily measured, thus giving an indication of the useful work being done by the engine.

An absorption dynamometer which may be attached to the engine crankshaft or flywheel absorbs the power in such a manner that it may be measured. The energy absorbed is converted into heat by friction and radiated to the surroundings or carried away by circulating cold water. One of the earliest forms developed is the *Prony brake* after Gaspard de Prony,¹ a French mathematician and engineer, who invented it about 1820. Several forms of water brakes² which are arranged to maintain, automatically, steady uniform loads are in use. Some of the water brakes are the reverse of centrifugal pumps, churning the water violently, and absorb very large quantities of energy per unit of size and weight.

¹ See *Annales de Chimie et de Physique*, Vol. 19, 1821.

² Carpenter and Diederichs, *Experimental Engineering*.

Geo. I. Alden, Transactions A.S.M.E., vol. 11, p. 958, for Alden absorption dynamometer.

Osborne Reynolds, Philosophical Transactions of the Royal Society, 1897, for description of a water brake.

F. Ratscher, description of a water brake designed by Prof. Johannes Stumpf, Zeitschrift der Verein deutscher Ingenieure, 1907, p. 607.

The Prony brake, in its simplest form, consists of two timbers hollowed out to fit the opposite sides of a shaft or flywheel of an engine. The timbers are squeezed onto the wheel, by tightening on two bolts as shown in Fig. 143, to produce the desired frictional load.

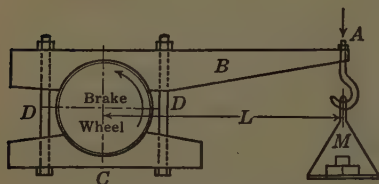


FIG. 143.—PRONY BRAKE

In measuring power by means of a brake, it makes no difference in the result whether the force measured at *A*, Fig. 143, is held in a fixed position and the brake wheel rotated, or whether the brake wheel is

fixed and the brake force rotated about it in a circle whose radius is *L*. The same quantity of work will be done in both cases provided the force is the same and the effective distance traveled per minute is the same. Therefore

$$(387) \quad \text{D. hp.} = \frac{\text{force in lbs.} \times \text{distance in ft. per min.}}{33\,000},$$

or

$$(388) \quad \text{D. hp.} = \frac{F \times 2 \pi L N}{33\,000},$$

in which

D. hp. = developed horsepower of the engine,

F = the force measured in lbs. at *A*,

L = radius of the brake arm in ft.,

N = r.p.m.

Then the friction horsepower is found by difference as indicated below:

$$(389) \quad \text{F. hp.} = \text{I. hp.} - \text{D. hp.}$$

The friction horsepower of an engine is distributed approximately as shown in Table 53.

Test No. 1 was made with a 6 × 12 straight-line balanced-valve engine. Test No. 2 was made with a straight-line engine with unbalanced valve.

The general laws of friction indicate, and test results demonstrate, that at constant speed the friction horsepower varies but

TABLE 53

DISTRIBUTION OF FRICTION LOSSES IN STEAM ENGINES ¹

TEST No.	(1)	(2)
Main bearings : per cent	47.0	35.4
Piston and rod per cent	33.0	25.0
Valve and rod per cent	2.5	26.4
Crank pin per cent	6.8	5.1
Cross head and wrist pin per cent	5.4	4.1
Eccentric strap per cent	5.3	4.0
Total per cent	100.0	100.0

little as the load on the engine changes. However, there is a tendency for the friction horsepower to increase as the engine output increases. The friction horsepower of an engine usually lies between 4 per cent and 15 per cent of the full-load indicated horsepower; the larger, better, and more expensive engines having the smaller friction loss.

In the days of Watt and continuing down to about 1850, it was customary to express steam-engine performance in terms of bushels of coal or pounds of coal burned in the boiler furnace per horsepower hour. Since the bushel might be a heaping, a rounded, or a struck bushel, and the cubic content of a bushel was variable anyhow, this was a very unsatisfactory method of measuring performance results. In the days of Professor Rankine and continuing down to about 1890, it was customary to state steam-engine performances in terms of pounds of coal burned per I. hp. per hour. The fuel consumption as determined by tests varied from about 2 to 8 pounds of coal per I. hp. per hour.² It was assumed that average American coals would produce about 12 500 B.t.u. per pound. Then it became the custom to express performance in pounds of steam supplied to the engine per I. hp. per hour. During the period from about 1890 to 1910 or thereabouts, the initial pressure and quality of steam did not vary greatly for

¹ See R. H. Thurston, *Friction and Lost Work in Machinery and Millwork*. Reprinted by permission of John Wiley & Sons, Publishers. This book is permanently out of print.

² See *Proceedings of the Institution of Civil Engineers* from 1836 to 1860.

engines in the same class. Hence the statement of the number of pounds of steam consumed per indicated horsepower hour or per developed horsepower hour is a more accurate basis of measurement than the older methods, and it was sufficiently accurate and satisfactory for many years. It continues to be used to a considerable extent for small units.

The steam rate is determined by testing the engine. The condensed exhaust steam, or *condensate*, and the fuel supplied to the boiler furnace are weighed. The conditions prevailing during the test, such as engine load, initial and exhaust-steam pressures, temperatures, and qualities are observed. The steam rate and other results are then computed. For modern engines, the steam rate varies from about 8 pounds per I. hp. per hour for large, correctly designed uniflow and compound engines supplied with superheated steam, to 40 or 50 pounds for small, cheaply built, simple engines supplied with wet steam and operated non-condensing.

In 1902 the American Society of Mechanical Engineers committee, appointed to standardize a system of testing steam engines, adopted the B.t.u. as the fundamental unit of energy by which to measure and compare engine performance. This unit had already been adopted by a similar committee appointed by the Institution of Civil Engineers of London. But the style of reporting engine performance in terms of steam rate still persists to a considerable extent. As time passes the *heat rate* must survive since it is scientific and accurate. It also permits direct comparisons to be made between the performance results of steam engines, steam turbines, and internal-combustion engines.

The temperature datum at which heat addition from fuel begins is *the temperature of the feed water entering the boiler*. Hence for a complete power-plant test, the A.S.M.E. testing codes specify that all heat quantities are to be computed above that level. In testing an engine unit separately from its boiler, the testing codes specify, and it is general practice, to assume the datum at an *ideal feed water temperature* equal to the temperature of the exhaust steam leaving the engine.

The equations and the symbols used in them for computing steam-engine-performance results are given on pages 367-8.

c_p = the mean specific heat of superheated steam at constant pressure,

e_c = thermal efficiency of the ideal Carnot cycle,

e_d = thermal efficiency of steam engine referred to D. hp.,

e_i = thermal efficiency of steam engine referred to I. hp.,

e_m = mechanical efficiency of steam engine,

e_R = thermal efficiency of the ideal Rankine engine,

e_r = efficiency ratio (actual engine to ideal Rankine engine),

H = total heat in 1 lb. of steam above 32° F., or enthalpy,

h = heat of the liquid for 1 lb. of water above 32° F.,

Q_d = heat supplied to an engine per D. hp. per hr.,

Q_i = heat supplied to an engine per I. hp. per hr.,

r = latent heat for 1 lb. of steam,

t = temperature of the boiling point of water, at the initial steam pressure,

t_s = temperature of superheated steam,

T = absolute temperature,

W_d = steam rate of an engine in lbs. per D. hp. per hr.,

W_i = steam rate of an engine in lbs. per I. hp. per hr.,

x = quality of the steam.

All heat quantities are measured in British thermal units. Efficiencies and steam qualities are expressed as decimal numbers.

Temperatures are all expressed in degrees Fahrenheit, t for thermometer readings and T for absolute temperatures.

The subscript 1 used with H , h , r , t , and x refer to values for the steam at the initial pressure. The subscript 2 refers to values for exhaust steam pressure.

The heat equivalent of one horsepower hour on an absolute basis is equal to

$$(390) \quad Q_a = \frac{33\,000 \times 60}{778.57} = 2543 \text{ B.t.u.}$$

For wet steam, for dry steam, and for superheated steam, the total heat for one pound above 32° F. is, respectively,

$$(391) \quad H = (xr + h),$$

$$(392) \quad H = (r + h) \quad (\text{found in the Steam Tables}),$$

and

$$(393) \quad H_s = r + h + c_p(t_s - t) \quad (\text{found in the Steam Tables}).$$

In general efficiency equals

$$(394) \quad e = \frac{\text{heat received} - \text{heat rejected}}{\text{heat received}},$$

$$(395) \quad e = \frac{\text{useful work expressed in heat units}}{\text{heat received}}.$$

As already developed,¹

$$(396) \quad e_e = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1},$$

and

$$(397) \quad e_R = \frac{H_1 - (x_2 r_2 + h_2)}{H_1 - h_2} = \frac{H_1 - H_2}{H_1 - h_2}.$$

In (397) the values of H_1 and H_2 may be read directly from the Mollier diagram for the test conditions. This makes it a very simple matter to determine the value of the Rankine cycle efficiency.

$$(398) \quad e_i = \frac{Q_a}{Q_i} = \frac{2543}{W_i(H_1 - h_2)}.$$

$$(399) \quad e_d = \frac{Q_a}{Q_d} = \frac{2543}{W_d(H_1 - h_2)}.$$

$$(400) \quad e_m = \frac{\text{D. hp.}}{\text{I. hp.}},$$

$$(401) \quad e_r = \frac{e_i}{e_R} \quad \text{or} \quad \frac{e_d}{e_R} \\ = \frac{2543}{W_i(H_1 - H_2)} \quad \text{or} \quad \frac{2543}{W_d(H_1 - H_2)}.$$

Since the Carnot engine is so very impractical, it is the general custom to consider the Rankine engine as the standard of perfection with which to compare all actual engines.

¹ See pp. 139-144 herein.

ILLUSTRATIVE PROBLEM ON STEAM-ENGINE PERFORMANCE

A simple high-speed engine with double-ported, balanced valve when operated condensing, gave the following test results at full load: $p_1 = 130$ lbs. per sq. in. gage; $t_1 = 355.5^\circ \text{ F.}$; $p_2 =$ vacuum of 25 in. of Hg.; $x_1 = 1.00$; I.hp. = 33.5; D.hp. = 30.5; barometer reading = 29.6 in. of Hg.; total weight of condensed steam = 730 lbs. per hr. Then

$$p_1 = 130 + (29.6 \times 0.491) = 144.5 \text{ lbs. per sq. in. abs.}$$

$$p_2 = (29.6 - 25.0)(0.491) = 2.26 \text{ lbs. per sq. in. abs.}$$

From the steam tables, t_2 corresponding to p_2 equals 130.7° F. , and $h_e = 98.6 \text{ B.t.u.}$

$$W_i = \frac{730}{33.5} = 21.80 \text{ lbs.}$$

$$W_d = \frac{730}{30.5} = 23.93 \text{ lbs.}$$

$$Q_i = 21.8 (1193 - 98) = 23\,870 \text{ B.t.u.}$$

$$Q_d = 23.93 (1193 - 98) = 26\,200 \text{ B.t.u.}$$

$$e_c = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{T_1} = \frac{355.5 - 130.7}{355.5 + 459.6} = 0.276$$

$$e_R = \frac{H_1 - H_2}{h_1 - h_e}$$

Taking values of H_1 and H_2 for the specific conditions of the test from the Mollier diagram, the result is

$$e_R = \frac{1193 - 918}{1193 - 98} = \frac{275}{1095} = 0.251.$$

$$e_i = \frac{2543}{23\,870} = 0.1066.$$

$$e_d = \frac{2543}{26\,200} = 0.0971.$$

$$e_m = \frac{30.5}{33.5} = 0.911.$$

$$e_r = \text{referred to I.hp.} = \frac{0.1066}{0.251} = 0.425 = 42.5\%.$$

$$e_r = \text{referred to D.hp.} = \frac{0.0971}{0.251} = 0.387 = 38.7\%.$$

Engine tests are usually carried out for a number of fractional loads on the engine. The results may then be presented in the

form of curves as shown in Figs. 144 and 145. Performance curves of this type furnish a convenient means of surveying at a glance the more important test results, and comparisons between the results of tests from various engines may be quickly made. It is seen by these figures that the steam-rate and heat-rate curves fall rapidly as the load increases up to a certain point and then they begin to rise. Usually steam engines are so rated that the minimum heat rate is obtained at about three-quarters of the full-

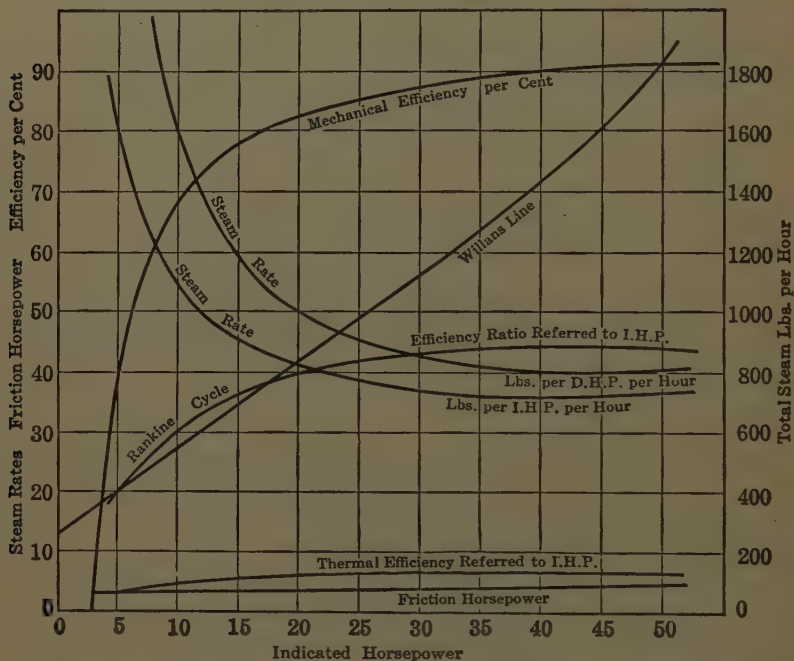


FIG. 144.—PERFORMANCE CURVES FOR SIMPLE NON-CONDENSING ENGINE

load rating. This will then permit of an overload of 25 to 50 percent without unduely increasing the steam and heat consumption. For various engine speeds, initial, and final pressures, the minimum steam rate is obtained at different loads. This explains very largely why the custom of rating engines in terms of cylinder dimensions instead of horsepower has grown up. The same cylinder may be given a number of different horsepower ratings depending upon the conditions of speed, initial steam pressure, and exhaust pressure.

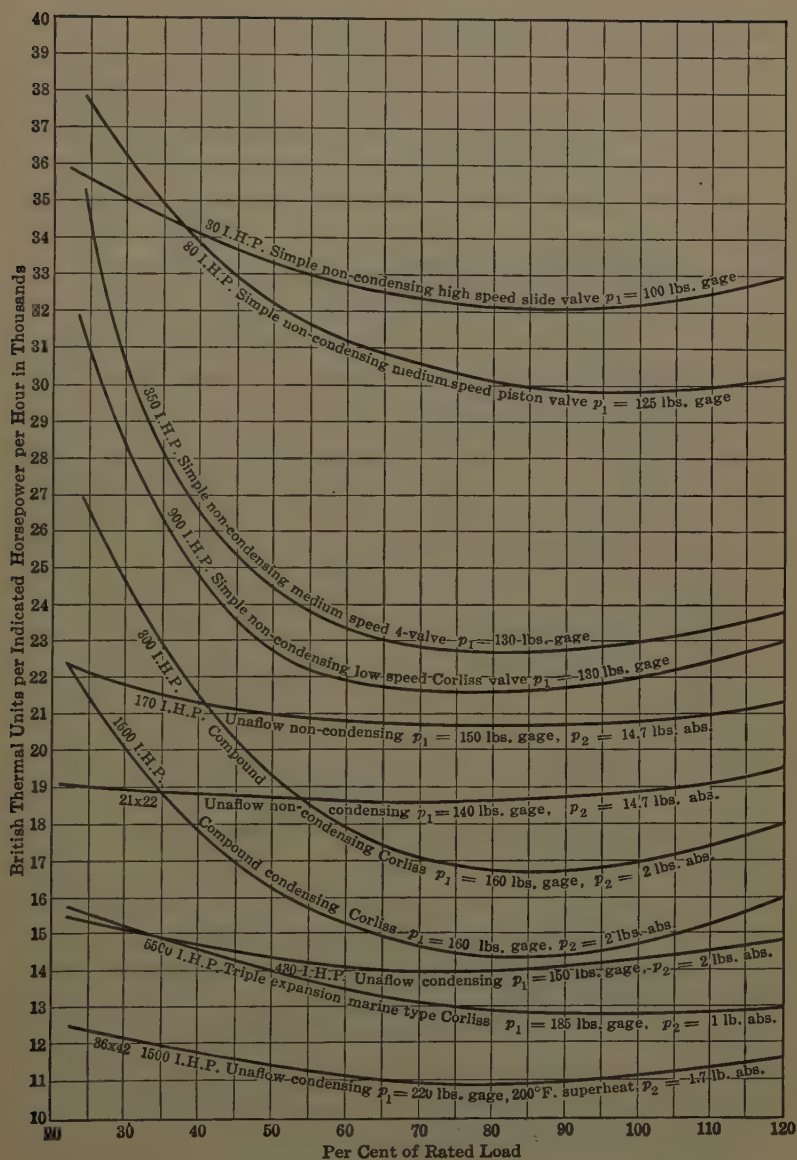


FIG. 145.—HEAT-RATE CURVES FOR STEAM ENGINES

175. Willans' Line and Other Performance Data. Peter W. Willans,¹ an English engine-builder, found by plotting as ordinates the total pounds of steam used by the engine per hour, and as abscissas the D. hp., the I. hp., or the kw. output, that the resulting curve is a straight line, or very nearly straight, except for overloads for which the curve bends upward rather sharply as indicated in Fig. 144. This curve is known as the *Willans line*, or the Willans law.

This law holds most strictly for engines controlled by the throttling governor. If the governor is of the variable cut-off type, the curve is generally somewhat convex to the x -axis. If the curve is exactly a straight line, two points, or one point and the slope, determine its location. Hence engine tests may be made for two load points; then the steam rates for all fractional loads may readily be computed. The curve seldom remains an exactly straight line for the very small loads; hence it is sufficiently accurate only for approximating steam-rate curves. For unaf flow engines, the curve is decidedly convex to the x -axis. The law is especially useful in estimating steam rates for steam turbines.

Engine builders generally prepare more or less elaborate curves of steam and heat rates for engines of various types and sizes, and for a variety of operating conditions which customers most frequently specify in their requests for new engines. By the use of such curves and tables, estimates as to size, steam rates, and prices of new engines may be quickly and cheaply prepared when the builders wish to submit bids on any new proposal. Representative data for two types of engines are shown in Tables 54 to 56 inclusive, and Fig. 145. Such tables and curves are generally the result of computations which have been verified by numerous shop and field tests of the various engine units. Such results are considered to be approximate only, and are modified as necessary in submitting competitive bids. In Tables 55 and 56, the engine capacities are based upon the maximum rated alternators to which they are direct connected.²

¹ See Minutes of the Proc. Inst. C. E. of London, vol. 114, pp. 2-112, 1893.

² Most modern alternating-current generators are given a maximum continuous rating, which, with class A insulation, gives a temperature rise not greater than 50° C. above the surrounding room temperature, as measured by a thermometer placed against the outside of the generator windings and held in position by lumps of sticky clay or putty placed over the thermometer bulbs.

The steam rates in Table 56 are per I. hp. per hour, and are based upon the assumption that approximately 75 per cent of the maximum rated alternator load at 80 per cent power factor is the most economical load.

The usual ratio of cylinder volumes in compound Corliss engines is given in Table 54.

TABLE 54
COMPOUND ENGINE CYLINDER RATIOS

P_1 Lbs. per Sq. In. Gage	Non- Condensing	Condensing
100	1 to 2.00	1 to 3.00
125	1 to 2.25	1 to 3.50
150	1 to 2.50	1 to 4.00
175	1 to 2.75	1 to 4.50

TABLE 55
DIMENSIONS AND WEIGHTS OF STEAM ENGINES

RATED I.H.P.	NO. OF CYLIN- DERS	R.P.M.	CYLINDER DIMENSIONS INCHES	WEIGHT OF ENGINE AND FLYWHEEL POUNDS	
				Total	Per I.hp.
UNAFLOW ENGINES					
80	1	257	10 x 12	14 000	175
160	1	257	12 x 16	23 000	144
320	1	200	18 x 20	40 000	125
475	1	200	20 x 22	50 000	105
790	1	150	26 x 30	80 000	101
1,260	1	150	28 x 36	102 000	81
14,000	4	75	36 x 60	1 000 000	71.5

COMPOUND CONDENSING CORLISS FOR PARALLEL OPERATION

320	2	150	11 x 22 x 30 at 34% C. O.	51 000	160
475	2	150	14 x 28 x 30 at 30% C. O.	68 500	144
790	2	120	18 x 36 x 36 at 33% C. O.	120 000	152
1,260	2	120	22 x 44 x 36 at 35% C. O.	174 000	138
1,580	2	120	22 x 44 x 42 at 40% C. O.	187 000	118

TABLE 56
COMPUTED STEAM RATES FOR STEAM ENGINES

RATED I. HP.	RATED R.P.M.	MECH. EFF.				EXHAUST PRESS. LBS. ABS.	P ₁ = 100 LBS. G. AT THROTTLE			
		Fractional Load					Fractional Load			
		$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{4}{4}$		$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{4}{4}$
UNAFLOW ENGINES										
45	300	.70	.84	.86	.90	20	29.0	27.2	26.8	27.0
						15	26.0	24.2	23.8	24.0
						2	20.8	19.2	18.6	19.5
80	257	.70	.84	.86	.90	20	28.0	26.4	26.0	26.2
						15	25.0	23.4	23.0	23.2
						2	19.7	18.4	18.0	18.4
160	257	.72	.85	.87	.91	20	27.0	25.7	25.4	25.6
						15	24.0	22.7	22.4	22.6
						2	18.6	17.7	17.5	17.5
320	200	.74	.86	.88	.92	20	25.5	24.4	24.0	24.5
						15	23.0	21.9	21.5	22.0
						2	17.8	16.5	15.8	16.3
475	200	.74	.86	.88	.92	20	25.3	23.9	23.6	24.0
						15	22.8	21.5	21.2	21.5
						2	16.2	15.3	15.0	15.3
790	150	.76	.87	.89	.93	20	24.6	23.1	22.8	22.9
						15	22.4	21.0	20.7	20.8
						2	14.9	13.9	13.9	14.1
1,260	150	.76	.87	.89	.93	20	24.0	22.7	22.2	22.3
						15	22.2	20.8	20.3	20.4
						2	14.3	13.1	13.1	13.3
CROSS-COMPOUND CORLISS ENGINES										
320	150	.74	.86	.89	.92	20	35.1	26.8	24.9	26.4
						15	32.4	25.3	23.3	23.8
						2	20.6	16.3	15.6	16.0
						20	34.1	26.4	24.7	26.2
475	150	.76	.87	.90	.93	15	31.6	24.9	23.1	23.6
						2	19.8	16.1	15.4	15.8
						20	33.3	26.0	24.4	26.0
						15	30.8	24.5	22.9	23.4
790	120	.78	.88	.92	.94	2	19.8	15.9	15.2	15.7
						20	32.7	25.7	24.1	25.8
						15	30.2	24.2	22.6	23.3
						2	19.1	15.6	15.0	15.5
1,260	120	.78	.88	.92	.94	20	32.5	25.5	24.0	25.7
						15	30.0	24.1	22.5	23.2
						2	19.0	15.5	15.0	15.5
						20	32.5	25.5	24.0	25.7
1,580	120	.78	.88	.92	.94	15	30.0	24.1	22.5	23.2
						2	19.0	15.5	15.0	15.5
						20	32.5	25.5	24.0	25.7

TABLE 56—Continued

COMPUTED STEAM RATES FOR STEAM ENGINES

$p_1 = 125$ LBS. G. AT THROTTLE				$p_1 = 150$ LBS. G. AT THROTTLE				$p_1 = 175$ LBS. G. AT THROTTLE			
Fractional Load				Fractional Load				Fractional Load			
$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{4}{4}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{4}{4}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{4}{4}$

UNAFLOW ENGINES

28.0	26.2	25.8	26.0	27.0	25.2	24.8	25.0	26.0	24.2	23.8	24.0
25.0	23.2	22.8	23.0	24.0	22.2	21.8	22.0	23.0	21.2	20.8	21.0
19.8	18.4	18.0	18.4	19.4	17.8	17.4	17.4	18.6	17.2	16.8	16.6
26.9	25.4	25.0	25.2	26.0	24.4	24.0	24.4	25.0	23.4	23.0	23.6
23.9	22.4	22.0	22.2	23.0	21.4	21.0	21.4	22.0	20.4	20.0	20.6
18.6	17.3	17.0	17.3	17.7	16.4	16.1	16.2	16.8	15.5	15.2	15.2
25.9	24.7	24.4	24.6	24.8	23.7	23.4	23.6	23.8	22.7	22.4	22.6
22.9	21.7	21.4	21.6	21.8	20.7	20.4	20.6	20.8	19.7	19.4	19.6
17.4	16.4	16.2	16.3	16.5	15.4	14.9	15.1	15.2	14.0	13.7	14.2
24.5	23.4	23.0	23.3	23.4	22.4	22.0	22.2	22.4	20.8	21.0	21.2
22.0	20.9	20.5	20.8	20.9	19.9	19.5	19.7	19.9	18.3	18.5	18.7
16.7	15.4	15.0	15.3	15.6	14.5	14.3	14.5	14.6	13.6	13.6	13.7
24.0	22.9	22.7	22.9	22.9	22.0	21.6	21.9	22.0	20.8	20.7	20.9
21.6	20.5	20.2	20.4	20.4	19.5	19.1	19.3	19.5	18.3	18.2	18.3
15.6	14.5	14.3	14.5	14.9	13.9	13.6	13.8	14.4	13.2	13.0	13.0
23.3	22.2	21.9	22.1	22.1	21.3	21.0	21.1	21.1	20.4	20.0	20.2
21.1	20.0	19.8	19.9	19.9	19.1	18.8	19.0	18.9	18.2	17.8	18.1
14.5	13.5	13.3	13.5	14.0	13.0	12.7	13.0	13.7	12.7	12.2	12.3
22.7	21.6	21.5	21.5	21.5	20.7	20.4	20.7	20.5	20.0	19.4	19.8
20.9	19.8	19.6	19.7	19.7	18.9	18.6	18.8	18.7	18.2	17.6	17.9
13.9	12.9	12.7	12.9	13.6	12.8	12.3	12.4	13.1	12.5	11.9	12.1

CROSS-COMPOUND CORLISS ENGINES

31.1	24.5	23.2	25.4	27.2	22.3	21.1	22.4	25.0	20.6	19.3	20.2
28.1	23.1	21.6	23.1	26.0	21.2	19.9	20.8	23.8	19.9	18.3	19.2
18.9	15.5	14.8	15.2	16.7	14.3	13.9	14.3	15.8	14.0	13.2	13.6
30.1	24.1	23.0	25.2	26.4	22.0	20.9	22.2	24.2	20.3	19.1	20.0
27.5	22.7	21.4	22.9	25.4	20.9	19.7	20.6	23.0	19.6	18.1	19.0
18.1	15.3	14.6	15.0	16.3	14.1	13.7	14.1	15.4	13.8	13.0	13.4
29.3	23.7	22.8	24.9	25.6	21.6	20.7	22.1	23.4	19.9	18.9	19.8
27.1	22.3	21.2	22.7	24.6	20.5	19.5	20.4	22.2	19.2	17.9	18.8
17.7	15.1	14.4	14.9	15.9	13.9	13.5	14.0	15.0	13.6	12.8	13.3
28.7	23.4	22.5	24.6	25.0	21.3	20.5	21.8	23.1	19.6	18.7	19.6
26.8	22.0	20.9	22.6	24.0	20.2	19.3	20.2	21.9	18.9	17.8	18.6
17.4	14.8	14.2	14.7	15.6	13.7	13.3	13.9	14.7	13.4	12.7	13.2
28.5	23.3	22.4	24.5	24.8	21.2	20.4	21.8	23.0	19.5	18.6	19.6
26.7	21.9	20.8	22.5	23.8	20.1	19.2	20.2	21.8	18.8	17.7	18.6
17.3	14.7	14.2	14.7	15.5	13.6	13.3	13.9	14.6	13.3	12.7	13.2

A comparison of the heat-rate data for unafLOW and Corliss engines in Table 56 and in Fig. 145 shows that the unafLOW engine is superior to both the compound and the triple-expansion Corliss engine. But some form of Corliss, usually the simple or compound style, is still preferred for many installations because of longer life and greater reliability due largely to its slow rotative speed.

In Table 55 are given the approximate dimensions and weights of unafLOW and compound engines equipped for parallel operation of alternating-current generators. These engines are designed for initial steam pressures ranging between 100 and 200 pounds per square inch gage. The unafLOW engine is consistently lighter than the compound engine for corresponding horsepowers, largely because of the higher speed; hence the required forces which operate to produce a given horsepower are correspondingly smaller, and the engine parts may be made smaller and lighter.

PROBLEMS

1. A 350 I.hp. non-condensing engine uses 3200 lbs. of steam per hour at quarter load and 8500 lbs. per hour at full load. The initial steam pressure is 125 lbs. per sq. in. gage, and the quality is 1.00; the exhaust pressure being 16 lbs. per sq. in. abs. Draw a Willans line and from it compute and plot the steam-rate curve.

2. Compute the Rankine-cycle efficiency and the efficiency ratio referred to the I.hp. and the B.t.u. supplied above the temperature of the exhaust in Problem 1.

3. Which is more economical on a B.t.u. basis, a simple non-condensing engine using 26 lbs. of dry and saturated steam per I.hp. per hour; initial pressure 120 lbs. per sq. in. gage; or a compound condensing engine using 12.5 lbs. of steam per I.hp. per hour, initial pressure 250 lbs. per sq. in. gage and superheated 200° F., exhaust pressure 4 in. of mercury absolute? Which is the more perfect engine? Give all results and computations therefor.

4. Explain in detail why the maximum attainable vacuum in a condenser is limited by the temperature of the available cooling water.

5. Derive equation (357) using methods suggested in the derivation of equation (355).

6. Draw a hypothetical indicator diagram, and use a suitable diagram factor to compute the expected full-load I.hp. of a 22 x 24 double-acting unafLOW engine similar to that shown in Fig. 95 when operating under the following conditions: $p_1 = 200$ lbs. per sq. in. gage; $p_2 = 2$ lbs. per sq. in. abs.; cut-off at 20 per cent of stroke; r.p.m. = 150; diameter of piston rod and tail rod = 3.75 in.

7. If the steam rate of the engine in Problem 6 is 11.00 lbs. per I.hp. per hour with the steam superheated 100° F. at the throttle, compute the values of

all the steam-engine efficiencies except the mechanical efficiency. The Mollier diagram may be used in solving this problem.

8. Under the conditions given in Problems 6 and 7, what would be the steam rate of the ideal Carnot engine; the ideal Rankine engine?

9. (a) Compute the probable I.hp. of a 16 x 20 double-acting engine operated at 275 r.p.m.; initial steam pressure 125 lbs. gage; cut-off at 25 per cent of stroke; expansion-curve law is $PV = C$; exhaust pressure is atmospheric. Use diagram factor of 0.78. (b) What is the piston speed of this engine?

10. A simple double-acting unafLOW engine is desired which will develop 250 I.hp. at full load under the following conditions: p_1 at throttle = 150 lbs. per sq. in. gage; initial superheat = 100° F.; exhaust pressure = 2 lbs. per sq. in. abs.; r.p.m. = 200; full-load cut-off at 20 per cent of stroke; expansion curve follows the $PV = C$ law. (a) Compute the hypothetical mean effective pressure, and draw the hypothetical indicator diagram. (b) Assume a suitable diagram factor, and determine suitable cylinder dimensions for a piston speed of 800 ft. per min.

11. A simple double-acting unafLOW engine 28" x 42" with 5½" piston and tail rods gave the following results on full-load test: clearance 4% (same for both ends); cut-off at 15% of stroke; release occurs at 95%; compression occurs during 85% of back stroke; p_1 at the throttle = 150 lbs. per sq. in. gage; p at cut-off = 140 lbs. per sq. in. gage; exhaust pressure = 2 lbs. per sq. in. abs.; x_1 at $p_1 = 1.00$; expansion and compression curves follow the law $PV = C$; barometer reading = 29.50 in. of Hg.; steam rate = 13.5 lbs. per I.hp. per hr.; r.p.m. = 150.

(a) Plot on cross-section paper an actual indicator diagram for one end of the cylinder. Assume that the diagrams are identical for both ends.

(b) Compute the mean effective pressure and the I.hp. from the diagram in (a). (Use a planimeter or count squares to determine the diagram area.)

(c) Compute the necessary results and plot the saturation curve on the sheet for (a).

(d) Compute the quality of the steam at a number of points during the expansion period, and draw a quality curve for which ordinates are qualities in per cent and abscissas are cylinder volumes in cubic feet.

12. Compute all engine efficiencies except the mechanical for the engine cited in Problem 11.

13. (a) On a second sheet of cross-section paper construct from the indicator diagram of Problem 11 the equivalent TN -diagram. Make the new diagram for one pound of steam. Plot absolute temperatures in degrees F. as ordinates extending the scale to absolute zero. (b) Draw in the lines of the steam dome and show the outline of the corresponding Rankine cycle. (c) Tabulate all important data and results.

14. If the exhaust from the engine in Problem 1 is used for heating purposes, what full-load steam rate should be charged to power? Give computations.

15. (a) Compute and plot thermal-efficiency curves referred to I.hp. for each of the four pressures and for the 790 I.hp. unafLOW-condensing engine data given in Table 56. (b) Repeat for the 1260 I.hp. compound-condensing Corliss engine data in the same table. (c) From the above curves for each engine, plot a curve for each exhaust pressure having as ordinates initial steam pressure and for abscissas thermal efficiency.

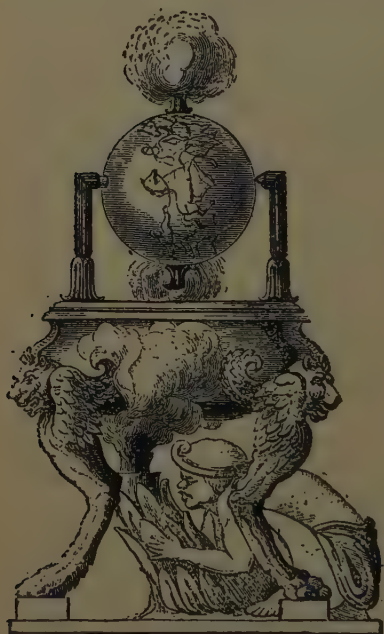
CHAPTER XV

STEAM TURBINES

176. Historical Sketch. Hero or Heron, who lived during the second century B.C., has been variously described as a Greek philosopher, physician, mathematician, and mechanic. But all agree that he was the first to describe in writing the whirling

æolipile or ball of Aeolus (Fig. 146) which, though only a toy, embodied the principles of the modern reaction steam turbine, and it was probably the first steam prime mover ever invented.

The fragments of Hero's writings which have been preserved describe a number of ingenious devices handed down to him from earlier times. Since he does not claim its invention, it is probable that the æolipile was known long before his day. It is uncertain that Hero's turbine ever ran at all, existing at that time only on paper and in men's minds. But it is supposed that it was used



(Smiles, *Lives of the Engineers*)

FIG. 146.—GREEK ÆOLIPILE

by the priests for driving so-called magical apparatus where invisible power and high speeds were desirable. "Another ingenious device was used to pour out libations. Upon the altar fire being kindled, the air confined in the interior of the machine became expanded, and pressing upon the surface of the liquid which it contained, forced it up a connecting pipe, and so into

the sacrificial cup. The libation was made and the people cried 'A Miracle'! But Hero knew the trick, and explained the arrangement by which it was accomplished: It forms the subject of his eleventh theorem."¹

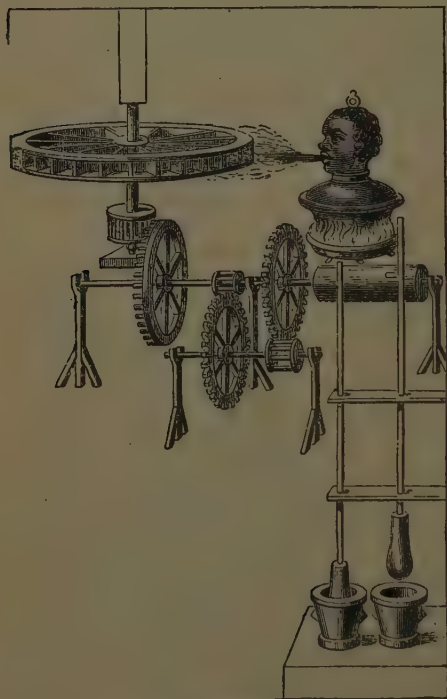
The æolipile consisted of a hollow metallic globe mounted on hollow trunions which communicated with an enclosed caldron of water underneath. The globe was provided with two or more projecting tubes closed on the ends but open on one side. The steam generated in the caldron was permitted to flow through the tubes into the atmosphere. The reactive force of the escaping steam caused the æolipile to spin on its axis "as if it were animated from within by a living spirit."

A copy of Hero's manuscript, *Spiritualia seu Pneumatica*, somehow escaped the fate of most ancient books in the libraries of Egypt, Greece, and Rome. A translation of it appeared in print in Bologna in 1547. Thus many copies became available for circulation throughout Europe. Other mechanisms were described in it, among them being arrangements for using the steam jet to accelerate combustion, several forms of steam boilers, and various hydraulic apparatus, for opening and closing temple doors, and for producing other mysterious temple effects at the psychological moment. It is said that eight editions of this book appeared in different languages within a century. This work probably excited much interest in the subject of mechanics and in the power of steam. But for a long time no advance beyond the æolipile was made; though many attempts were made to apply its rotary motion to useful work. In 1577 a German mechanic employed the æolipile to turn roasts before the fireplace instead of turnspit dogs, and it is supposed that he used it to rotate burnishing and reaming tools.

In 1629 Branca, an Italian architect, invented the impulse steam turbine as shown in Fig. 147. He used a steam jet issuing from the lips of a brazen head, and impinging upon vanes to drive a sort of stamp mill for pounding drugs. The pursed lips formed an elementary nozzle from which the steam jet issued at high velocity. Branca's turbine included all the elements of a modern

¹ Samuel Smiles, *Lives of the Engineers*, vol. 4, Chap. I.

impulse steam turbine except the enclosing case. But because Branca and the others at that time lacked the knowledge of the thermodynamic properties of expanding gases and vapors, the turbine did not become a successful prime mover until the latter part of the 19th century; though much effort was expended upon it and many patents taken out. Thermodynamic laws and knowledge had to be acquired by labored experimental develop-



(Smiles, *Lives of the Engineers*)

FIG. 147.—BRANCA IMPULSE TURBINE

ments by many investigators, and through the development of the reciprocating steam engine by Savery, Newcomen, and Watt before the turbine could become commercially important.

177. Parsons and His Reaction Turbine. Charles A. Parsons, fourth son of William Parsons, third Earl of Rosse, was born in 1854, and educated at St. Johns College, Cambridge. Parsons' first turbine was built about 1882 along the lines of Hero's engine with the idea of investigating the merits of the reaction type. The sphere was replaced by two hollow oval sectioned

arms from which the steam issued through jets turned tangentially to the plane of motion. The whole was enclosed in a cast-iron case which was connected to a condenser.

With an initial steam pressure of 100 lbs. per sq. in. and exhausting into a vacuum of 26 inches of mercury and at 5000 r.p.m., the turbine developed 20 brake horsepower with a steam rate of 40 pounds per horsepower per hour.

In 1884, encouraged by his initial experiment, he built a 10-horsepower, 18 000 r.p.m., multistage turbine consisting of a drum on which were mounted 30 rows of outwardly projecting brass blades alternating with corresponding rows of fixed blades projecting inwardly from the casing. The steam was admitted at the middle of the casing and flowed axially toward the exhaust at each end. It embodied all the essential features of the modern Parsons type of reaction turbine. This turbine was the pioneer in a long line of commercially successful reaction-type turbines. It was operated for several years in Gateshead, on Tyne, England, supplying electric current for the manufacture of incandescent lamps. It is now in the South Kensington Museum.

Many of the details of this first turbine are now obsolete, but the fundamental principle of operation is retained in the most modern and largest units now built (see Figs. 192 to 196 inclusive). To Sir Charles A. Parsons is undoubtedly due the credit for having built the first commercially successful steam turbine which marked the beginning of an enormous industry in the generation and distribution of power.

178. De Laval and His Turbines. Dr. Carl Gustaf De Laval was born at Bläsenborgs, Dalarna, Sweden, in 1845. His college training was obtained at the Institute of Technology at Upsala, from which place he received the degree of Ph.D. in 1872. Until 1877, he conducted a general contracting and engineering business. In 1878, he received a patent for a cream skimmer, and the next year invented the centrifugal cream-separator. Thereafter he devoted himself exclusively to his inventions and their promotion.

His first steam turbine, which he built in 1882 to drive his high-speed cream separator, operated on the reaction principle. He was granted patents on this turbine in 1883. But it appeared unsuited to his particular needs. For one thing, the steam consumption was very large in the small units required for the cream-separator drive. Hence he turned his attention to the impulse type. In 1887 after having investigated both types, he decided to adopt the one first invented by Branca, and he was the first to develop this type into a commercially successful machine. His

most important patent, which was granted in 1894, covered the application of the diverging nozzle to direct the steam jet onto the rotating buckets as shown by Fig. 148. This turbine was first manufactured in this country in 1896 by the De Laval Turbine Company of Trenton, N. J. A modern form of it is shown in section by Fig. 149.

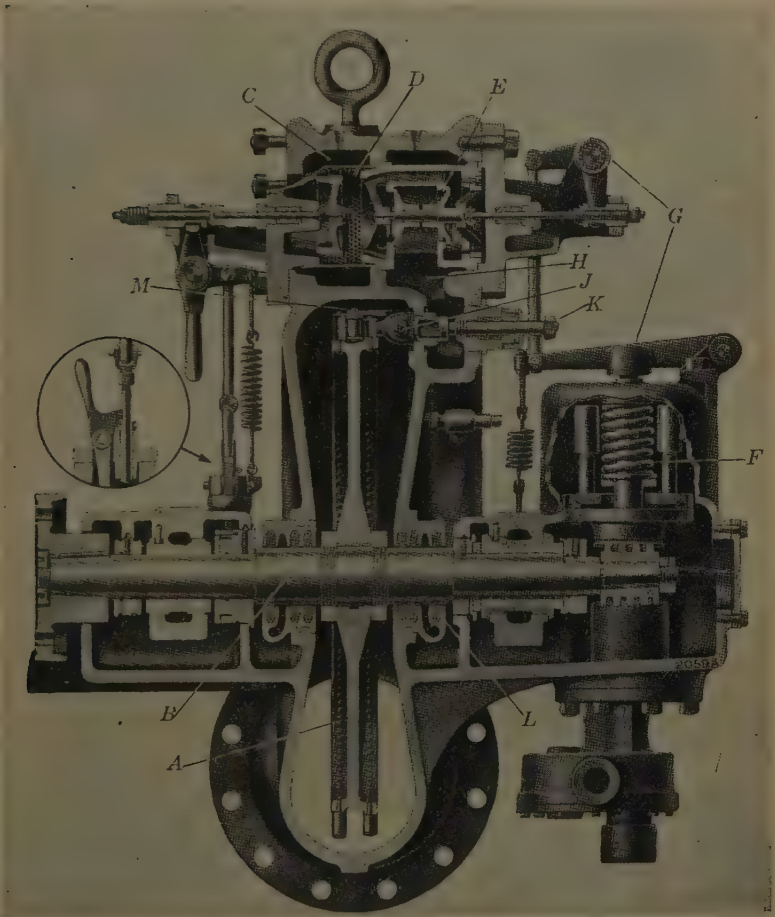


(De Laval Steam Turbine Co.)

FIG. 148.—DE LAVAL BUCKET WHEEL AND EXPANDING NOZZLE

The velocity-stage turbine shown in Fig. 149 has a single rotating wheel *A* which is provided with two rows of buckets (ordinarily called double-bucket row). The wheel is made of forged steel accurately machined and dynamically balanced. It is keyed to the shaft and further secured in place by lock-nuts. The wheel revolves in steam at exhaust pressure, and as the

pressure is practically equal on the two sides of the wheel there is no appreciable axial thrust. The buckets of nickel steel are assembled on the shroud ring in groups, as shown in Fig. 150 (c), and are then riveted to the wheel rim.



(De Laval Steam Turbine Co.)

FIG. 149.—DE LAVAL SINGLE-STAGE TURBINE

The steam from the boiler enters the steam chest *C* (Fig. 149) by a flanged connection not shown. It passes through a heavy monel-metal strainer *D* on its way to the balanced-piston throttle-valve *E* operated in a removable cage by the speed-governor *F*.

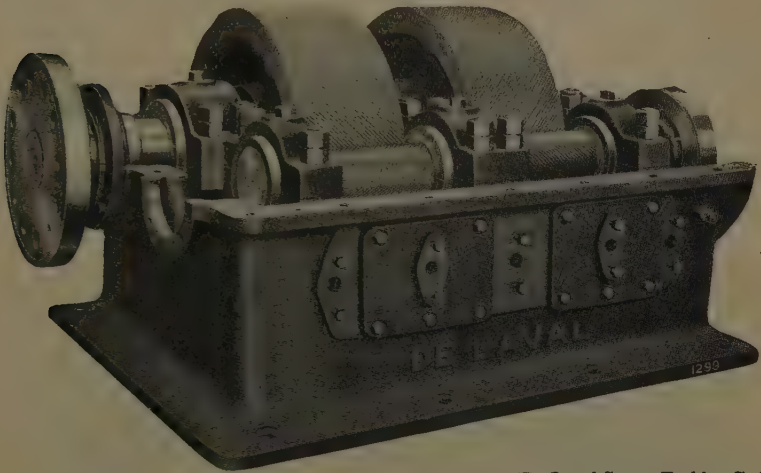
acting through spring-loaded levers *G*. The throttling governor controls the steam pressure in the nozzle chamber *H*. The expanding nozzles, Fig. 150 (*b*) and *J*, Fig. 149, carrying the stationary intermediate guide vanes, Fig. 150 (*b*) and *M* Fig. 149, are attached to the wheel-chamber wall. Hand valves *K* are employed to control the number of nozzles open, thus preventing excessive and uneconomical throttling of the steam by the governor for the small fractional loads. Carbon-ring shaft packings *L*, which prevent excessive leakage of steam, are supported in the shaft housing. They are provided with a steam seal when the turbine is to be operated under vacuum. The steam seal effectively prevents leakage of air into the wheel chamber.



FIG. 150.—NOZZLES AND BUCKETS FOR DE LAVAL TURBINE

The wheel and shaft are mounted in a split casing, which is rigidly supported near the center line by a single heavy pedestal. This arrangement permits the casing to expand due to temperature changes without affecting the alignment of the bearing and gland housings. The two-velocity-stage style of De Laval turbine illustrated in Fig. 149 is built for capacities ranging from 1 horsepower to 800 horsepower. They are largely employed to drive, direct-connected or through reduction gearing, Fig. 151, electrical generators, centrifugal pumps, blowers, centrifugal air compressors, and other rotating machinery.

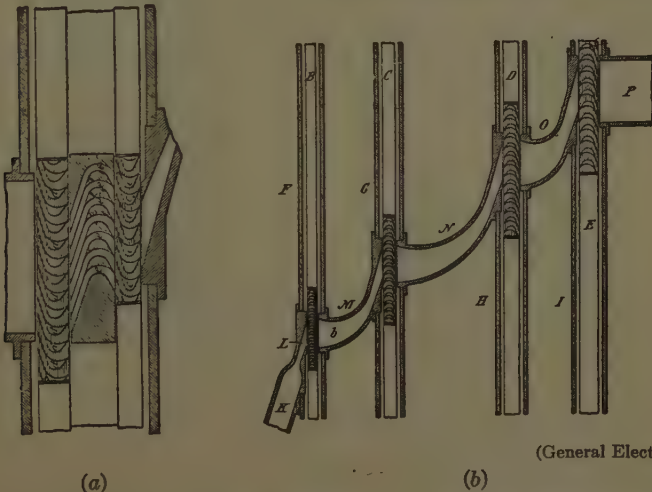
De Laval designed the first double-helical reduction gears for driving low-speed machinery in 1889. The high efficiency and



(De Laval Steam Turbine Co.)

FIG. 151.—HELICAL REDUCTION GEAR WITH COVER REMOVED

long life of these gears, which have been adopted by other turbine manufacturers, is due to the fact that they were correctly designed



(General Electric Co.)

FIG. 152.—ORIGINAL CURTIS PATENTS U. S. Nos. 566 968 and 566 969

and accurately cut. Experience has shown that these gears, when made of ample proportions, correctly cut from proper

materials, and maintained in true alignment by suitably constructed housings, will transmit small or large quantities of power practically indefinitely at sustained efficiencies of 97 to 99 per cent.

179. Curtis and His Turbines. Charles Gordon Curtis was born in Boston, April 29, 1860. His technical education was

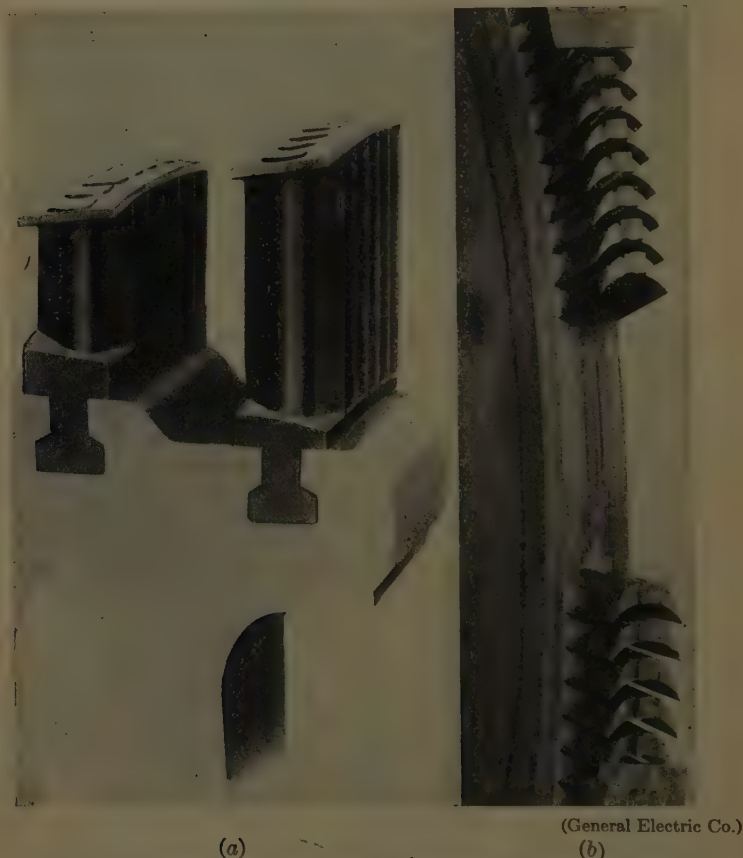


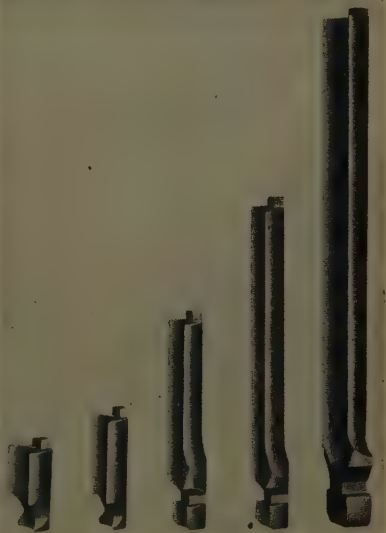
FIG. 153.—METHOD OF ATTACHING BUCKETS AND SHROUD RING

obtained at Columbia University, from which place he received the following degrees: C.E., 1881; LL.B., 1883; and M.S., 1907. He practiced for eight years as a patent lawyer. He organized two electric corporations, The C. and C. Electric Motor Com-

pany, and the Curtis Electric Manufacturing Company, of which he was president.

In 1896, he was granted three United States patents on fluid turbines. The first patent dealt with a new turbine-speed-control device which varied the nozzle area open to the passage of steam in accordance with the load demand. This device was never used extensively. The second patent provided for conversion of energy of the steam issuing from the nozzle at very high velocity in a series of moving buckets mounted on one wheel, as illustrated in Figs. 152 (a), 153 (a), and 155. This arrangement is known as *velocity stages*. By this expedient the high-velocity steam can be efficiently utilized in buckets having low peripheral speeds. This was a result impossible of realization in the De Laval turbine. His third patent covered an arrangement of a series of nozzles and bucket wheels in series as shown in Figs. 152 (b) and 201, known as *pressure stages*. By this expedient, the pressure drop was divided up into a number of steps, each step constituting

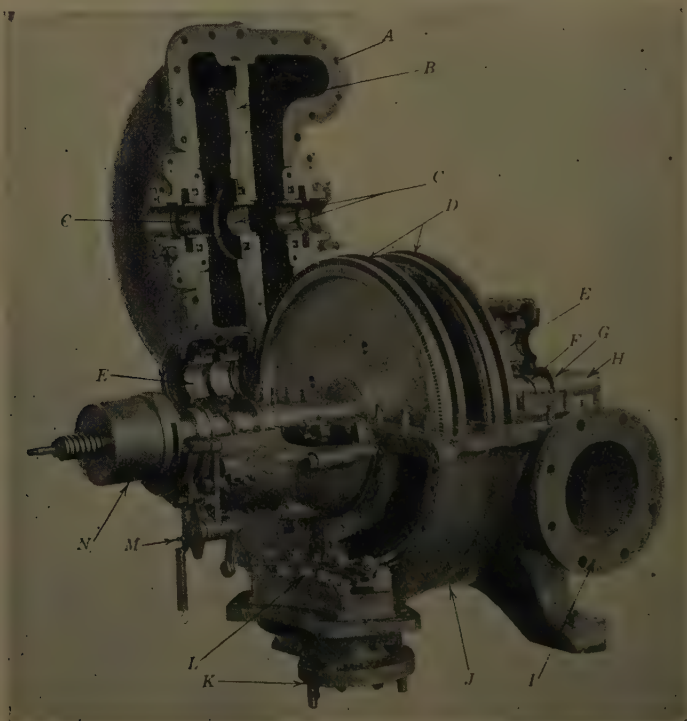
a single De Laval turbine. By employing a sufficient number of pressure stages, the velocity of the steam impinging upon the buckets can be held to values small enough to give good efficiencies of energy conversion. Curtis also combined both velocity stages and pressure stages in the same machine as shown by Fig. 155. In some of the earlier turbines, manufactured by the General Electric Company, as many as four rows of buckets were mounted upon each wheel. But several rows of buckets on one wheel do not give the high efficiency desired. The general



(General Electric Co.)

FIG. 154.—TYPICAL IMPULSE BUCKETS

practice in modern turbines is to employ not more than two velocity stages (two rows of moving buckets) per pressure stage as illustrated in Figs. 152 (*a*), 153, and 155. About 1900, he sold the land rights for his turbine patents to the General Electric Company. During the years 1900 to 1905, this corporation spent over \$2 000 000 developing the turbine, with the result



(General Electric Co.)

FIG. 155.—CURTIS TWO-STAGE MECHANICAL-DRIVE TURBINE

A, top half of shell; *B*, diaphragm between stages; *C*, labyrinth packing; *D*, double-bucket-row wheel; *E*, top half of main bearing; *F*, oil ring; *G*, oil deflector rings; *H*, steel shaft; *I*, exhaust flange; *J*, bottom half of shell; *K*, steam-chest flange; *L*, nozzle plate; *M*, emergency over-speed stop; *N*, centrifugal governor case.

that now the Curtis and Parsons type turbines are the principal heat engines employed in large central power stations in the United States. Both types are also much used in marine propulsion, being used in the United States Navy and in the British navy to propel the largest battleships as well as smaller craft.

180. Other Turbines.¹ From time to time since 1900, other forms of turbine, all of the impulse type, have been invented, patented, and placed on the market. Among them may be mentioned the Terry, Kerr, and Sturtevant turbines. Fig. 156 shows the Terry impulse turbine, in which velocity-staging effects are



(The Terry Steam Turbine Co.)

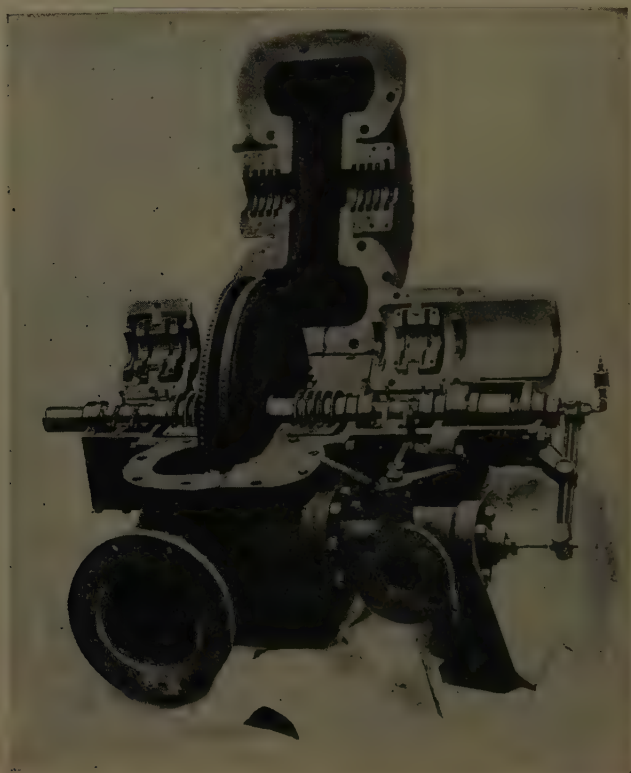
FIG. 156.—TERRY IMPULSE TURBINE

produced in a single row of buckets by directing the steam flow through a spiral-shaped passage leading back into the moving bucket row of the single wheel several times. The efficiency realized in this form of turbine is not as high as is possible in the Curtis arrangement of a series of wheels carrying double-bucket rows (Fig. 155). But the Terry design of turbine possesses the

¹ See Geo. A. Orrok, *Small Steam Turbines*, Transactions A.S.M.E., vol. 31, 1909.

advantage of being a cheaper machine to build than one having a double or triple bucket-row wheel. However, this advantage of cheapness is of first importance only in the smaller machines.

Modern forms of Kerr impulse turbines made by the Elliott Company, Pittsburgh, Pa., are illustrated in Figs. 157 to 161 inclusive. The single-stage machine contains a double row of



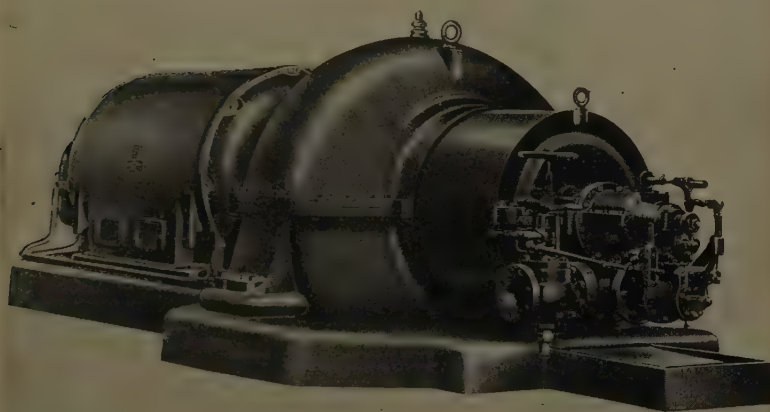
(Elliott Co.)

FIG. 157.—KERR SINGLE-STAGE TURBINE

moving buckets. Each row is mounted on a separate wheel forged from special steel. The Kerr wheels are keyed and shrunk on the shaft, and are further secured in place by means of shrink-rings which fit into recesses on each side of the wheel as shown in Fig. 159.

The buckets are drop forgings made from nickel steel. The

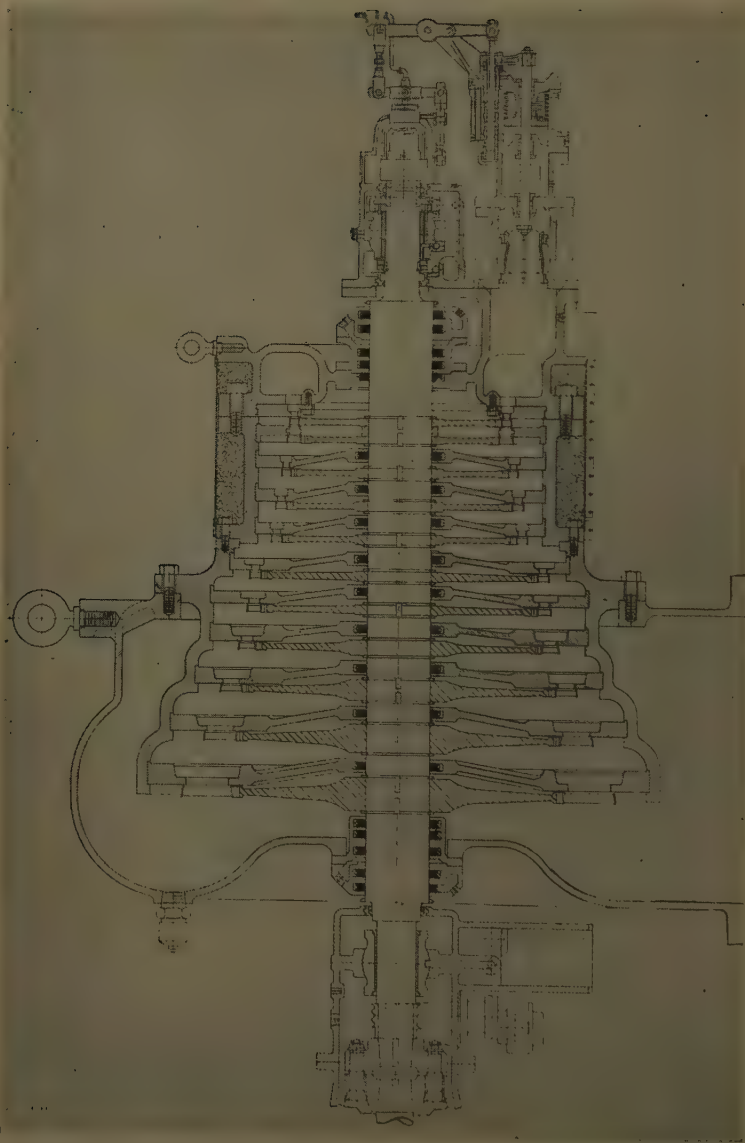
forging hammers impart to them a hard glazed surface of natural metallic oxide which is very resistant to the erosive action of steam. The buckets are unsymmetric in form. That is, the inlet and exit angles are not equal, the latter being the smaller in order to reduce the loss due to the velocity of the leaving steam. The bucket heights on their exit side are also greater than on the inlet side, to give increased area of flow to compensate for the loss of velocity as the steam passes through the buckets, and to compensate for the reduction in the width of the jets leaving due to the sharper exit angles.



(Elliott Co.)

FIG. 158.—KERR CONDENSING TURBINE-GENERATOR

In many industries low-pressure steam is desired for heating buildings and for industrial processes. This is especially true in textile, paper, and woodworking establishments. Bleeder or extraction turbines are very convenient sources of this low-pressure steam, provided the quantity required for industrial purposes is less than that required to operate the turbine. In a bleeder turbine the entire steam supply passes through the higher stages of the turbine, generating power. Then at the proper point in the turbine to give the desired pressure, sufficient steam for the industrial-heating demand is extracted, or bled from the



(Elliott Co.)

FIG. 159.—LONGITUDINAL SECTION OF KERR CONDENSING TURBINE

turbine, and the remainder of the steam continues through the lower stages. The employment of bleeder turbines proves to be very successful and economical in many industrial plants. Figs. 160 and 161 show the automatic mechanism for controlling the extraction of steam from Kerr bleeder turbines.

It consists of a specially designed slide valve controlled by a spring-loaded diaphragm. A small pipe from the heating system

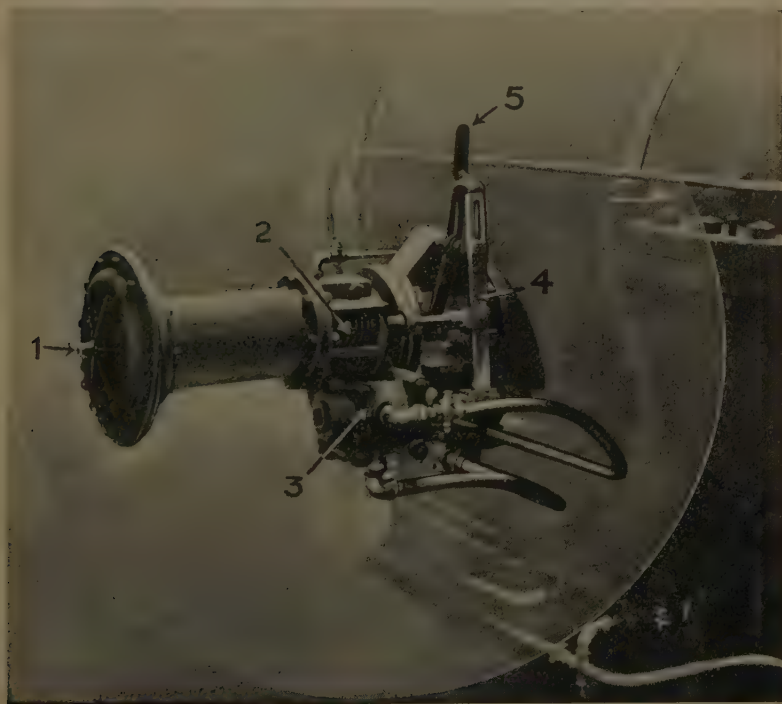
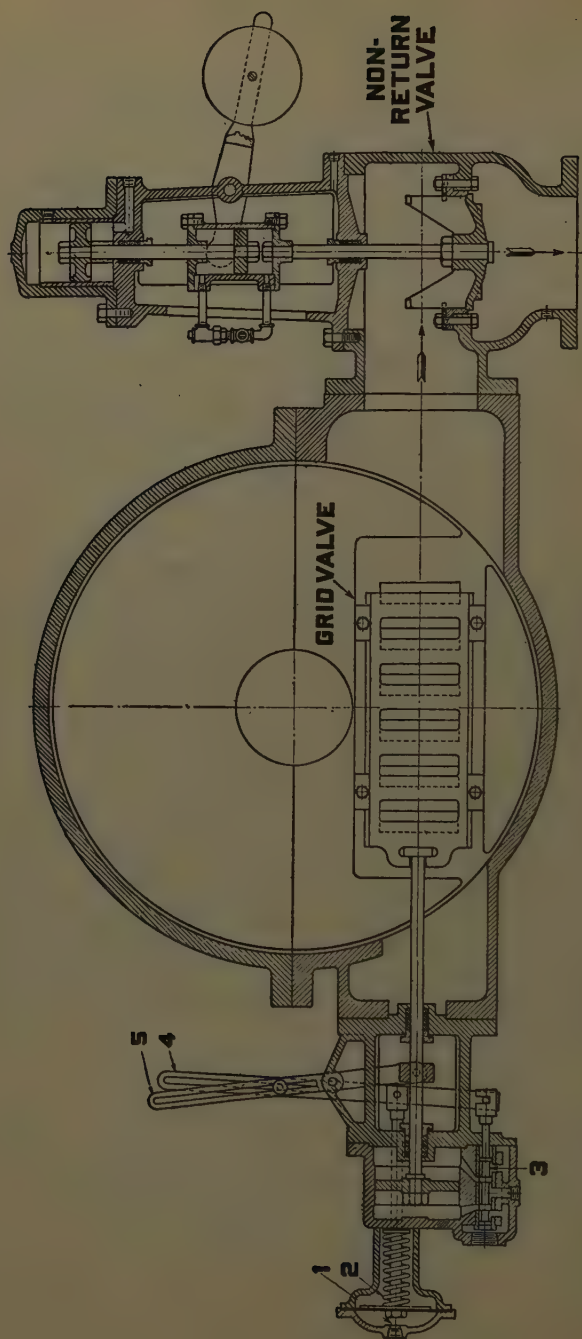


FIG. 160.—BLEEDER-TURBINE VALVE GEAR

(Elliott Co.)

is connected to one side of a flexible diaphragm at 1, Fig. 160. When the pressure of the heating system rises too high, the regulating diaphragm is deflected to the right. This motion is opposed by the coiled spring 2. This movement is transmitted to lever 4, which is pivoted at one end to a second lever 5, and attached at the other end to the pilot valve 3. Movement of the pilot valve admits oil under pressure to the main oil cylinder.



(Elliott Co.)

FIG. 161.—BLEEDER-TURBINE GRID-VALVE MECHANISM AND NON-RETURN VALVE

This moves the main piston and piston rod which is attached directly to the grid valve, Fig. 161. This movement opens additional ports in the grid valve, permitting more steam to pass into the lower stages of the turbine, thereby reducing the pressure in the heating system to normal. The linkage of levers is such that the movement of the piston rod returns the pilot valve to its neutral position. When the pressure of the heating system falls too low, the coil spring forces the diaphragm to the left and



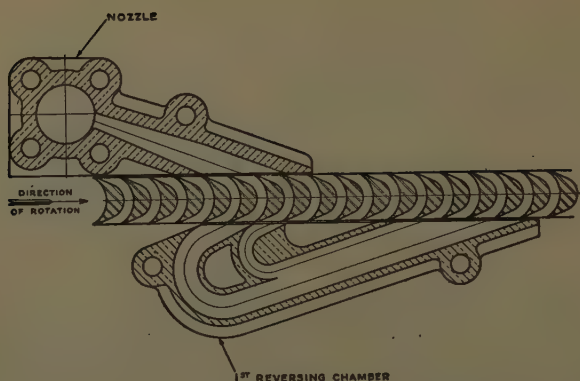
(Westinghouse Electric and Manufacturing Co.)

FIG. 162.—WHEEL AND NOZZLES OF WESTINGHOUSE MECHANICAL DRIVE TURBINE

A, nozzle; B, bucket wheel; C, reversing chamber; D, water-seal runner; E, oil rings.

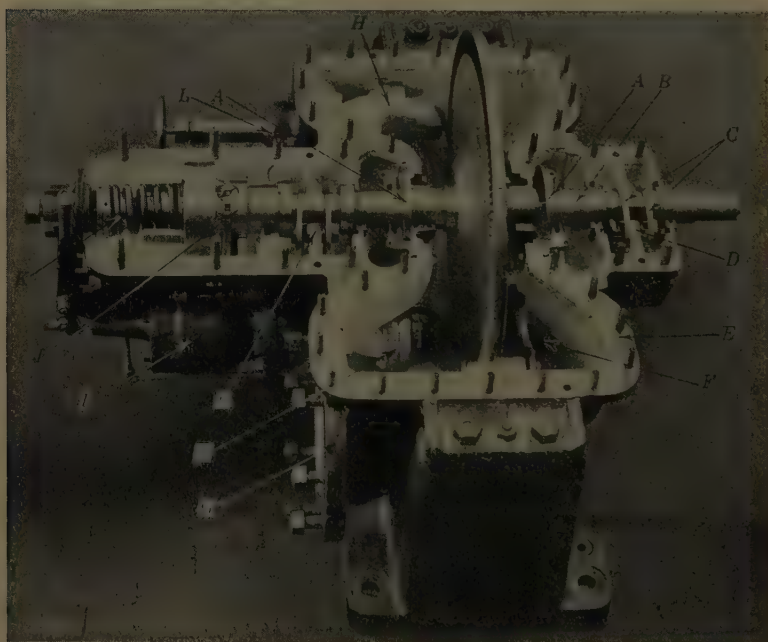
the slide-valve closes or partially closes the ports. This allows less steam to pass on through the turbine and more steam is delivered to the heating system. These functions of the mechanism together with the speed-governor of the turbine are carried out rapidly enough to maintain practically constant turbine speed and normal pressure in the heating system.

The mechanism includes a non-return valve installed on the



(Westinghouse Electric and Manufacturing Co.)

FIG. 163.—SECTIONAL VIEW OF NOZZLE AND BUCKETS OF WESTINGHOUSE TURBINE



(Westinghouse Electric and Manufacturing Co.)

FIG. 164.—WESTINGHOUSE MECHANICAL DRIVE TURBINE

A, water seal runner; B, forged steel shaft; C, oil rings; D, main bearings; E, cast iron shell; F, reversing chambers; G, exhaust flange; H, nozzle blocks; I, steam chest; J, emergency stop; K, centrifugal governor; L, over-load hand valve.

bleeder outlet to protect the turbine against overspeeding due to possible reversal of steam flow. In the case of reversal of flow back into the turbine, the steam will lift the non-return valve to its seat and stop the flow. If for any reason this valve should fail to close, and the turbine speed increase more than 10 per cent above normal, an automatic over-speed governor closes the main steam-valve. Additional protection is supplied in the form of an over-speed vacuum breaker mounted on the exhaust end of condensing turbines. These last two features are not shown.

Other forms of automatic bleeder valves, varied as to details of construction and operation, are used by the other turbine manufacturers when the installation requires them.

The Westinghouse Electric and Manufacturing Company have accomplished velocity-staging on a single-row impulse wheel by employing the re-entry principle of the Terry turbine as illustrated by Figs. 162, 163, and 164.¹ The development of some form of impulse turbine was necessary on the part of the Westinghouse Company if they wished to compete successfully in the small turbine field, because the construction cost of small reaction turbines is too high and the heat efficiency too low.

181. Impulse and Reaction. The performance of steam turbines depends, as the names of the two general types imply, upon the manner in which the driving forces are developed in the rotor; namely, by direct action or impulse forces as illustrated in the De Laval, Curtis, and Kerr machines, and upon reactive forces as illustrated in Figs. 165 and 166.

In these figures, the tanks are assumed to be kept full of water, and they are mounted upon frictionless rollers. The tanks are provided with nozzles from which jets of water issue and impinge upon blocks D and D' . The blocks are also mounted upon frictionless rollers. In each case the blocks will be forced away from the tanks, unless restrained, due to the direct forces b and b' developed when the jets of water strike them, and the momentum of the jets is transferred to the blocks. This direct action of the jets is called an *impulse force* or simply *impulse*. As the jets

¹ See Power, p. 502, April 12, 1912, for a description of the Kienast turbine, which was an early form, if not the earliest, of the re-entry type.

issue from the nozzles, they exert oppositely-directed forces a and a' upon the tanks in accordance with Newton's third law, which is, *Action and reaction are equal and opposite in direction*. These **reaction forces**, or simply **reactions**, are equal to the respective direct forces or impulses. They will cause the tanks, unless restrained, to move away from the blocks with increasing velocity in accordance with Newton's second law which may be stated thus: *Mass times acceleration equals force*.

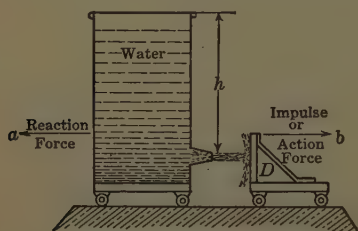


FIG. 165.—IMPULSE AND REACTION

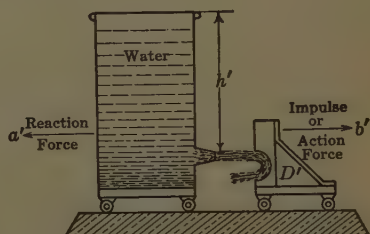


FIG. 166.—IMPULSE AND REACTION

In Fig. 166, the block has a curved surface which, in the ideal case, would reverse the direction of flow of the jet through 180 degrees with the velocity of flow undiminished. The block is then acted upon by two equal forces simultaneously, both tending to move it to the right. When the water particles strike the block an impulse force is exerted, and as they leave a reactive force or reaction is exerted upon it. If both jets are identical in size and form, and if the water leaves them with equal velocities and densities, then the total force acting upon block D' is just twice as great as the force acting on block D .

The development of impulse and reaction forces in steam-turbine buckets is illustrated in Figs. 167, 168, and 169. A bucket wheel and expanding steam nozzle directing the flow of steam into the buckets are shown in Fig. 167. If the wheel were held stationary, the steam would leave the buckets in a direction parallel to the axis of the shaft. Hence the reaction force of the steam jet upon the buckets tending to rotate the wheel would be zero. But there would be a small reaction force axially toward the left, as indicated by the arrows on the right of the wheel. The force developed by the steam entering the buckets would be

divided into a tangential component tending to rotate the wheel and an axial thrust, which would be partially balanced by the reaction of the steam leaving the buckets.

Fig. 168 shows the buckets with surfaces concave toward the entering steam, thus the direction of flow of the steam is nearly reversed as it passes through the buckets. With this arrangement the impulse of the jet as it enters the buckets is divided into tangential and axial components and the steam leaving the buckets also produces both tangential and axial forces due to reaction. The two axial forces act in opposite directions and

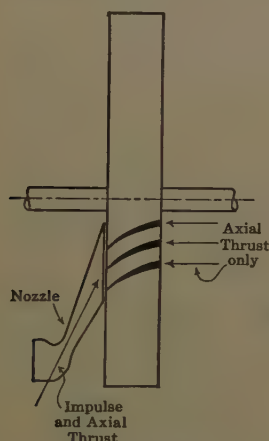


FIG. 167.—SIMPLE
IMPULSE

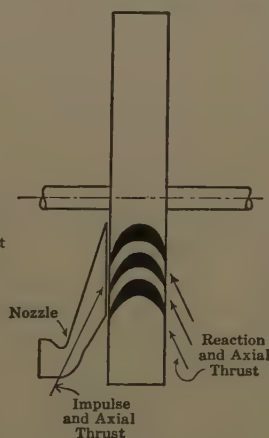


FIG. 168.—IMPULSE
AND REACTION

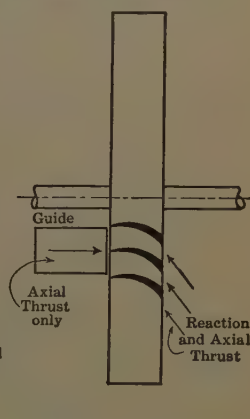


FIG. 169.—SIMPLE
REACTION

nearly balance so there is minimum net axial thrust on the wheel.

In Fig. 169 the buckets are curved oppositely to those in Fig. 167, in which case only the reactive force of the leaving steam tends to rotate the wheel. The arrows shown in these figures to the right of the wheels are intended to indicate only the direction in which the reactive forces operate on the wheel, and not their magnitudes.

The effective turning effort per pound of steam (provided the initial and final steam pressures and qualities are the same for all cases) on the buckets of Fig. 168, is nearly twice as great as in either of the other arrangements due to the reaction of the leaving steam. Also in Figs. 167 and 168, nearly all the steam

expansion takes place in the nozzles with the result that the steam enters the buckets at high velocity. In Fig. 169 there is no nozzle in which steam expansion takes place, in the strict sense, but only a guide to direct the steam into the buckets. Consequently the steam enters the buckets at relatively low velocity. Most of the expansion and acceleration due to the pressure drop occurs in the buckets. Hence the place in which the major part of the steam expansion and acceleration occur serves to distinguish between impulse and reaction types of turbine. However, some slight expansion and acceleration of steam does occur in the buckets of impulse turbines and reactive force is developed; also the steam in a reaction turbine enters the buckets at a velocity higher than that of the buckets and the buckets are curved in both directions as shown in Fig. 185. Hence there is also some direct action in reaction turbines. Thus it may be stated that there are no pure impulse turbines, and no pure reaction turbines. However, in all impulse turbines direct action greatly predominates, and in all reaction turbines reaction forces predominate.

182. Absolute and Relative Velocities. A body moving through space may have many velocities depending upon the reference datum from which the velocity is observed. Suppose, for example, that a baseball thrown by a pitcher standing on the earth is observed by the pitcher, by a person riding in an automobile on the earth, and by an inhabitant of Mars (the supposition that the inhabitant of Mars could see and observe the ball is also understood). The velocity at which the ball would appear to be moving through space would be different to each observer. There could be as many velocities as observers provided each were stationed upon different planets or other stellar bodies traveling through space.

So far as is known, there is no absolutely stationary point or body in space from which true absolute velocities could be measured. But by common agreement the term "*absolute velocity*" in steam-turbine design is applied to the velocity at which a body moves through space as measured from some fixed reference point on the earth by an observer stationary on the

earth, yet moving through space with the earth's speed. All velocities measured from some other datum are called *relative velocities*.

In order to make specific the illustration of the baseball pitcher and the automobile, assume that the observer in the automobile is travelling along a highway at a velocity of 50 feet per second relative to the earth. A baseball pitcher standing on the earth 100 feet away from the highway throws the ball to the observer in the car at a velocity, relative to the earth, of 200 feet per second. These two velocities are absolute velocities. To the observer traveling in the automobile the ball appears to be approaching him at a smaller velocity and from a direction different than appears to be the case to the pitcher stationed on the earth. This state of affairs is represented graphically and to scale in Fig. 170 by vectors where

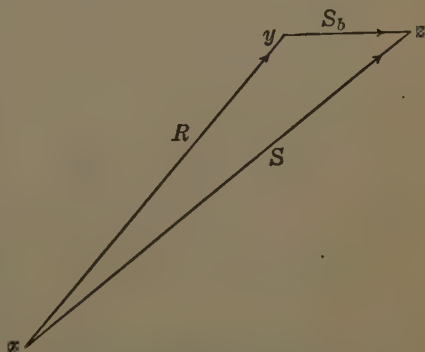


FIG. 170.—VELOCITY DIAGRAM OF BASEBALL

S_b = the absolute velocity of the automobile in magnitude and direction,

S = the absolute velocity and direction in which the pitcher threw the ball,

R = the relative velocity of the ball, in magnitude and direction as it appears to the observer in the automobile.

The same relationships exist for steam issuing at high velocity from a fixed nozzle and impinging upon a bucket moving at lower velocity on a turbine wheel. In Fig. 171 steam is represented by the arrow as flowing from the nozzle with an absolute velocity of S feet per second. A turbine bucket is represented as passing the nozzle exit with an absolute, tangential velocity of S_b feet per second. The directions and magnitudes of these

velocities are represented by the direction and length of the respective arrows as they would appear to an observer stationed on the earth. An observer stationed on the bucket and moving with it through space as the wheel revolves would see the steam approach from the direction and with the velocity indicated by the vector R in Fig. 172, which is known as relative velocity.

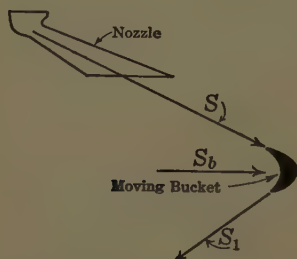


FIG. 171.—STEAM VELOCITIES IN NOZZLE AND BUCKET

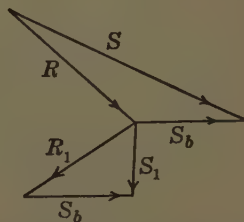


FIG. 172.—VELOCITY DIAGRAMS FOR STEAM AND MOVING BUCKET

Steam in an impulse turbine leaves the nozzle at velocities varying all the way from 200 or 300 feet per second up to a velocity of 4000 feet per second. The exit velocity from the bucket is from 15 to 20 per cent less due to turbulence and friction of the steam flowing between the buckets.

183. Power and Efficiency of Impulse Turbines. By theoretical mechanics it has been shown that the kinetic energy, in foot-pounds, of a rapidly moving mass is

$$(402) \quad KE = \frac{MS^2}{2g},$$

and

$$(403) \quad \text{hp.} = \frac{KE}{550} = \frac{MS^2}{1100g} = \frac{MS^2}{35\,390},$$

in which

g = acceleration due to gravity = 32.174 or approximately 32.2 in this latitude,

hp. = horsepower,

KE = kinetic energy in ft.-lbs. per sec.,

M = weight of steam or other moving body in lbs. per sec.,

$\frac{M}{g}$ = Mass of the moving body in British gravitational units,

S = absolute velocity of the moving body in feet per sec.

For example, in an imaginary impulse turbine, 5 lbs. of steam per second impinge upon the buckets at an initial velocity of 4000 feet per second, and the steam leaves the buckets at a velocity of 1000 feet per second. Neglecting friction losses, what would be the horsepower developed? What would be the efficiency?

$$\text{hp.} = \frac{5 (4000^2 - 1000^2)}{35\,390} = 2119.$$

The horsepower possible if the efficiency were 100 per cent is

$$(\text{hp.})_{100} = \frac{5 \times 4000^2}{35\,390} = 2260.$$

Since efficiency = $e = \frac{\text{result}}{\text{effort}}$,

then

$$e = \frac{2119}{2260} = 0.938 = 93.8 \text{ per cent.}$$

184. Maximum Power and Efficiency of a Single-Stage Impulse Turbine. This is an ideal case in which there is no frictional loss in relative velocity of the steam as it passes through the buckets mounted on the rim of the rotating wheel. The velocity relations in magnitude and direction of the steam and buckets are represented by the vectors in Fig. 173. The following symbols will be used in the necessary equations.

S = absolute velocity of the steam leaving the nozzle, feet per sec.,

S_b = absolute linear velocity of the buckets, ft. per sec.,

R = relative velocity of the steam entering the buckets, ft. per sec.,

S_1 = absolute velocity of the steam leaving the bucket, ft. per sec.,

R_1 = relative velocity of the steam leaving the bucket, ft. per sec.,

α = angle between the axis of the nozzle and the tangential direction of motion of the bucket to an observer on the earth,

β = angle between the direction of motion of the steam entering the bucket and the tangential motion of the bucket, to an observer on the bucket.

γ = angle between the direction of motion of the steam leaving the bucket and the motion of the bucket to an observer on the bucket.

The value of angle α varies from 12 to 25 degrees. However, the angle most used is about 20 degrees. The angles β and γ are sometimes made equal by the construction of the bucket.

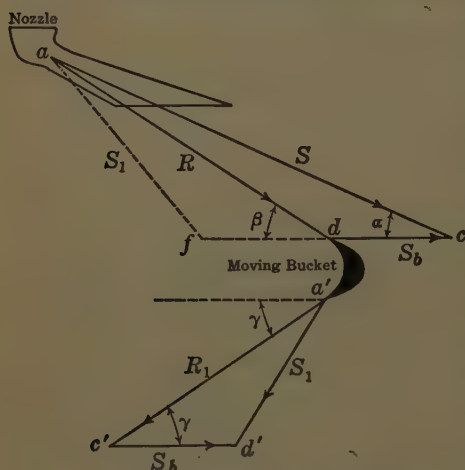


FIG. 173.—DIAGRAMS FOR STEAM AND MOVING BUCKET

This eliminates or greatly reduces axial thrust on the turbine wheel. The values of β and γ are usually between 24 and 30 degrees in a single-bucket row. In a double row of moving buckets the values are usually, first row 24 to 40 degrees, and in the second row 35 to 50 degrees. In any case angles are chosen which produce the best nozzle-and-bucket efficiencies

for the prevailing conditions of steam pressure and temperature.

In the triangles acd and $a'c'd'$, $R = R_1$, since there is no velocity loss due to friction as the steam passes through the buckets. S_b is the same in both triangles. Then the triangle $a'c'd'$ may be transposed so that $a'c'$ coincides with ad of triangle acd . Since angle $\beta = \gamma$, cdf will be a straight line whose length equals $2 S_b$. Since the values of S and S_b may be readily calculated, the law of cosines may be employed to solve for S_1 in triangle acf thus,

$$(404) \quad S_1^2 = S^2 + (2 S_b)^2 - 2 S(2 S_b) \cos \alpha,$$

and

$$(405) \quad S^2 - S_1^2 = 4 SS_b \cos \alpha - 4 S^2 b.$$

Now $S^2 - S_1^2$ is a measure of the energy, in foot-pounds, given up by the steam as it passes through the moving buckets. The amount of this energy will be maximum when (405) has a maximum. A function has a maximum value when its first derivative equals zero and its second derivative has a negative value. For greater convenience in manipulation, let

$$(406) \quad \begin{cases} S^2 - S_1^2 = y \\ S_b = x \\ S \cos \alpha = k. \end{cases}$$

The quantity k is a constant for a given turbine and fixed initial steam pressure. Substituting (406) in (405) one obtains

$$(407) \quad y = 4 kx - 4 x^2,$$

then

$$(408) \quad \frac{dy}{dx} = 4k - 8x,$$

and

$$(409) \quad \frac{d^2y}{dx^2} = -8.$$

Equating (408) to zero and solving, one finds

$$(410) \quad 8x - 4k = 0.$$

and

$$(411) \quad x = \frac{k}{2}.$$

Therefore (409) and (411) indicate that the energy given up by the steam to the buckets is maximum when

$$(412) \quad S_b = \frac{S \cos \alpha}{2},$$

or when

$$(413) \quad \frac{S_b}{S} = \frac{\cos \alpha}{2}.$$

When the nozzle angle α is made 20 degrees as is the case in many impulse turbines, the maximum energy available and consequently the *maximum heat efficiency of the turbine* is obtained when

$$(414) \quad \frac{S_b}{S} = \frac{0.94}{2} = 0.47$$

In order to realize this maximum efficiency, the steam must leave the moving buckets with zero relative velocity; that is, S_1 , in triangle $a'c'd'$ (Fig. 173) must be at right angles to the direction of motion of the buckets, or S_1 must be at right angles to S_b . This most desirable result is obtained when the absolute velocity of the bucket is 0.47 of the absolute velocity of the steam leaving a 20-degree nozzle.

185. Flow of Fluids through Nozzles. Assume a gas or a vapor flowing in a nozzle as shown in Fig. 174. By making a study of all the forms of energy at any two sections as indicated

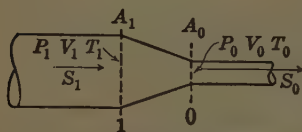


FIG. 174.—FLOW OF FLUIDS

by 1 and 0, equations may be written from which the velocity of the fluid at either section may be computed. Symbols will be used to represent the quantities involved as previously explained or as given in the table below.

Subscript 1 refers the value of quantities at the section marked 1 in Fig. 174, and the subscript o refers to the quantities at the section marked o . The following symbols will be used:

- A = area of the section in sq. ft.,
- c_n = nozzle-velocity coefficient,
- E = total energy in B.t.u.,
- H = total B.t.u. above 32° F. in one lb. of steam,
- H_t = B.t.u. used in increasing temperature only,
- k = ratio of specific heats,
- I = internal work in B.t.u.,
- M = weight of fluid flowing, lbs. per sec.,
- P = pressure lbs. per sq. ft.,
- Q = heat change in B.t.u. between sections 1 and 0,
- r = ratio of pressures = $\frac{P_0}{P_1}$,

S = velocity of fluid, ft. per sec.,

v = specific volume, cu. ft. per lb.,

V = total volume flowing, cu. ft. per sec.,

$\frac{W}{J}$ = external work done between 1 and 0, B.t.u.;

ρ = total internal energy, B.t.u.

$$(415) \quad \rho = H_t + I.$$

In order to obtain simple equations certain assumptions are necessary as follows:

1. That the fluid molecules move in smooth stream lines without friction or eddies.
2. That the fluid is perfectly elastic, filling the conduit or nozzle at all sections.
3. That the motion is steady and uniform; *i.e.* at a given section of the conduit the pressure, temperature, specific volume, velocity, etc., are constant and uniform at every point in the section under consideration.
4. That at every section the same weight of fluid passes in unit time.

Equation (416), which is known as the *continuity equation*, may be written directly from the 4th assumption.

$$(416) \quad \frac{V_1}{v_1} = \frac{V_0}{v_0} = \frac{A_1 S_1}{v_1} = \frac{A_0 S_0}{v_0} = M.$$

In the general case, the total energy E_1 at section 1 may not be equal to E_0 . The changes in E may occur in two ways: first, heat energy may be added or given out equal to $\pm Q$ B.t.u.; second, external work may be done by or upon the fluid equal to $\pm W/J$ B.t.u.

The only evidence of external work at A_1 is the work done in forcing the fluid across the section which equals $P_1 v_1$ foot-pounds for each pound of fluid. At A_0 the external work is likewise equal to $P_0 v_0$ per pound of fluid. Hence the net quantity of work stored in the fluid in flowing from section A_1 to section A_0 is

$$(417) \quad \pm \frac{W}{J} = \frac{P_1 v_1 - P_0 v_0}{J}.$$

The total energy at A_1 plus any energy received by the fluid in passing from A_1 to A_0 must equal the total energy of the fluid at A_0 . Hence

$$(418) \quad E_1 \pm Q \pm \frac{W}{J} = E_0.$$

Equation (418) may be stated in words as follows. The sum of the kinetic and potential energies at A_1 plus the energies added or lost between A_1 and A_0 equals the sum of the kinetic and potential energies at A_0 . For unit weight of fluid, this relationship may be restated as in (419):

$$(419) \quad \frac{S_1^2}{2gJ} + \rho_1 + \frac{P_1 v_1}{J} \pm Q = \frac{S_0^2}{2gJ} + \rho_0 + \frac{P_0 v_0}{J}.$$

In the case of a nozzle which is usually very short, where no heat energy is added or lost, the flow is adiabatic and Q is zero. When transposed and rearranged (419) becomes

$$(420) \quad \frac{S_0^2 - S_1^2}{2gJ} = \left(\rho_1 + \frac{P_1 v_1}{J} \right) - \left(\rho_0 + \frac{P_0 v_0}{J} \right).$$

But for steam

$$\rho + \frac{Pv}{J} = H.$$

Hence (420) for steam becomes

$$(421) \quad \frac{S_0^2 - S_1^2}{2gJ} = H_1 - H_0,$$

and

$$(422) \quad S_0 = \sqrt{2gJ(H_1 - H_0) + S_1^2}.$$

In steam-turbine practice, the value of the velocity S_1 entering the nozzle is usually negligibly small; hence when the numerical values of g and J are introduced (422) becomes

$$(423) \quad S_0 = 223.8 \sqrt{H_1 - H_0}.$$

Equation (423) is used to compute the theoretical or *spouting velocity* of steam discharged from a nozzle. The value of S_0 is independent of the weight of steam flowing per second and the

size of the nozzle, the only condition being that the flow shall be truly adiabatic. Steam-turbine nozzles are so short and the velocity is so high that there is practically no time for the flowing steam to lose heat by radiation; so the expansion is almost exactly adiabatic. Since the expansion is practically adiabatic or with constant entropy, the values of H_1 and H_0 in (423) may be most conveniently read from the Mollier diagram as explained on page 358. But there is some slight loss in velocity due to friction of the steam molecules against the nozzle wall and internally among the molecules themselves; thus the actual velocity at A_0 is somewhat smaller than S_0 as computed from (423). This is taken care of by introducing a nozzle-velocity coefficient thus,

$$(424) \quad S = c_n S_0,$$

in which S is the actual absolute velocity of the steam as it leaves the nozzle and enters a row of moving buckets. The value of the velocity coefficient c_n usually lies between 0.95 and 0.98. For smooth well-designed nozzles 0.97 is a value much used.¹

Equation (421) may be restated as follows:

$$(425) \quad \frac{S_1^2}{2g} + JH_1 = \frac{S_0^2}{2g} + JH_0 = K.$$

This means that for adiabatic flow of fluids the sum of the velocity energy and thermal energy is a constant at all sections of a tube or nozzle. If a term were added to each member of equation (425) for the energy change above datum, the result would be the well-known Bernoulli's theorem in a new form. In the case of steam pipes and nozzles the change in datum head is usually negligible, and for that condition the extra terms are not required to make (425) a complete statement of Bernoulli's theorem.

186. Weight of Steam Discharge by Nozzles. Napier's formula, equation (217), is sufficiently accurate to use in calculating the weight of steam flowing through a calorimeter orifice, but it is not sufficiently accurate to use in steam-turbine nozzle design. Napier's formula gives results which may be in error

¹ See Transactions A.S.M.E., 1926, pp. 33-64.

from 2 to 5 per cent, especially for superheated steam, or if the initial pressure greatly exceeds 100 lbs.

The weight of steam flowing through a nozzle may be computed from the continuity equation (416) and the expression for velocity in (425) thus,

$$(426) \quad M = \frac{A_0 S_0}{v_0} = \frac{223.8 A_0 \sqrt{H_1 - H_0}}{v_0}.$$

It is not always convenient or even possible to determine the values of H_0 and v_0 .

Consider the case of flow of a perfect gas through an orifice whose sectional area toward the exit of the orifice decreases along smooth, bounding curves as in Fig. 175. The cross-sectional area of the orifice exit is then a minimum. Also consider the flow to be frictionless and adiabatic. Actually, it is nearly so because the length of path through the orifice and its exposed surface is so small that the time required for a molecule of the

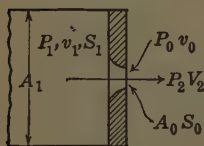


FIG. 175

gas to pass through the orifice is extremely short. (The usual gas velocities in orifices of this type are between 200 and 3000 feet per second and the nozzle is only 2 or 3 inches long.)

The symbols employed below are as explained in § 185. The quantities P_1 , S_1 , A_1 , and v_1 are measured in the pipe upstream from the orifice and P_0 , S_0 , A_0 , and v_0 are measured in the orifice throat or exit. It will be assumed that the area A_1 is so large that the effect of S_1 may be neglected without appreciable error. Since the fluid remains a perfect gas, there will be no change in the value of the internal energy as the fluid flows from area A_1 to area A_0 due to addition or extraction of heat energy. Hence equation (420) may be written thus,

$$(427) \quad \frac{S_0^2}{2g} = (P_1 v_1 - P_0 v_0).$$

The work of delivering the gas along the pipe to the orifice and increasing the velocity through the orifice must be supplied by the energy already stored in the gas, and is represented by the

area $abcd$ on the PV -plane in Fig. 176. Ideally, this work is equal to the work of compressing-and-delivering an equal quantity of a gas along an adiabatic by an ideal air compressor as shown on page 97. But in the case of the orifice the work is done by the gas and not upon it. For one pound

$$\int_{P_0}^{P_1} v dP,$$

hence equation (427) may be written

$$(428) \quad \frac{S_0^2}{2g} = - \int_{P_0}^{P_1} v dP.$$

The minus sign preceding the integral term simply indicates that the work to produce flow is furnished by the gas. The relation between

P and v during adiabatic expansion of the gas through the orifice is

$$Pv^k = K.$$

Therefore (428) becomes

$$(429) \quad \frac{S_0^2}{2g} = \frac{k}{k-1} (P_1 v_1 - P_0 v_0),$$

and

$$(430) \quad S_0^2 = \frac{2gkP_1 v_1}{k-1} \left(1 - \frac{P_0 v_0}{P_1 v_1} \right).$$

From the equation of state

$$\frac{P_0 v_0}{P_1 v_1} = \frac{T_0}{T_1},$$

and for adiabatic flow

$$\frac{T_0}{T_1} = \left(\frac{P_0}{P_1} \right)^{(k-1)/k}.$$

Hence (430) becomes

$$(431) \quad S_0^2 = \frac{2gkP_1 v_1}{k-1} \left[1 - \left(\frac{P_0}{P_1} \right)^{(k-1)/k} \right].$$

Placing the value of S_0 from (431) in (426) and reducing, one

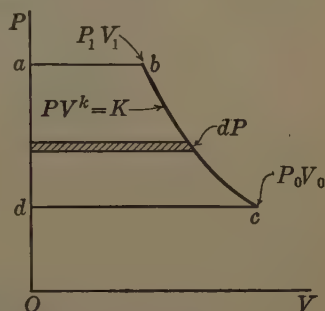


FIG. 176.—WORK OF ADIABATIC FLOW

obtains

$$(432) \quad M = \frac{A_0}{v_0} \sqrt{\frac{2 g k P_1 v_1}{k-1} \left[1 - \left(\frac{P_0}{P_1} \right)^{(k-1)/k} \right]}.$$

Since for adiabatic change

$$\frac{v_0}{v_1} = \left(\frac{P_1}{P_0} \right)^{1/k}, \quad \text{or} \quad v_0 = v_1 \left(\frac{P_1}{P_0} \right)^{1/k},$$

and from the equation of state

$$(433) \quad \frac{P_1}{v_1} = \frac{P_1^2}{R T_1},$$

and let

$$\frac{P_0}{P_1} = r$$

equation (432) becomes¹

$$(434) \quad M = \frac{A_0 P_1}{\sqrt{T_1}} \sqrt{\frac{2 g k}{R(k-1)}} (r^{2/k} - r^{(k+1)/k}).$$

The value of M is then expressed in quantities which are more readily measured than if equation (428) were employed.

The value of M becomes maximum when

$$r^{2/k} - r^{(k+1)/k} \quad \text{is maximum.}$$

Let

$$(435) \quad y = r^{2/k} - r^{(k+1)/k}.$$

Then y is maximum when

$$(436) \quad \frac{dy}{dr} = \frac{2}{k} r^{(2-k)/k} - \frac{k+1}{k} r^{1/k} = 0,$$

or when

$$(437) \quad r = \frac{P_0}{P_1} = \left(\frac{2}{k+1} \right)^{k/(k-1)}.$$

For air the value of k is about 1.403, and it turns out that maximum flow through an orifice occurs when

$$P_0 = 0.527 P_1.$$

¹Equations (431) and (434) were first developed by Saint Venant and Wantzel. See Comptes Rendus de l'Académie des Sciences, vol. 9 (1839).

Substituting (437) in (432), combining terms, and reducing, one obtains

$$(438) \quad M_{\max} = A_0 \left(\frac{2}{k+1} \right)^{1/(k-1)} \sqrt{\frac{2 g k}{k+1} \frac{P_1^2}{R T_1}}.$$

For air $R = 53.36$; $k = 1.4034$; and $g = 32.17$; then (438) becomes

$$(439) \quad M_{\max} = 0.531 P_1 \frac{A_0}{\sqrt{T_1}}.$$

Equation (439) is known as Fliegner's formula.¹ For dry and saturated steam the value of k is approximately 1.135,² and the maximum flow occurs when

$$P_0 = 0.58 P_1.$$

The above demonstration proves that the velocity and discharge rate of fluids will increase with a reduction of the throat pressure P_0 until that pressure becomes nearly one-half of the initial pressure. This pressure P_0 which gives maximum velocity is known as the *critical flow pressure* to distinguish it from the critical pressure at which change of state occurs. It also follows that the value of pressure and volume just at the orifice exit must remain fixed at the critical-flow values upon lowering the pressure outside the orifice. In other words, while the pressure at the orifice exit remains fixed at the critical value, the pressure just outside the orifice may be much less. The explanation of this paradox is that the transmission of a pressure wave through a fluid requires time. The maximum velocity of flow occurs when it reaches the maximum velocity at which the pressure wave can be propagated through the fluid under consideration. By substituting (437) in (431) and reducing one sees that this velocity is the velocity of sound in the medium (see texts on Physics).

After the fluid escapes into the air or into a vacuum chamber where the pressure is much lower than the critical, it naturally

¹ See *Der Civilingenieur*, vol. 20, 1874, p. 14; and vol. 23, 1874, p. 443.

² For more complete information on values of k for steam see Marks' *Mechanical Engineers' Handbook*, p. 348.

expands to the lower pressure, and the kinetic energy available from this further expansion is reconverted into heat by internal friction and eddies. Hence an orifice is not a suitable device to develop all the available kinetic energy from a fluid flowing through it into a region where the pressure is below the critical. However if the orifice is extended beyond the throat with properly diverging walls, the entire pressure drop may be efficiently converted into velocity.

187. Napier's Formula. When the pressure P_0 , Fig. 175, is equal to the pressure of the region into which the fluid is discharged from an orifice or nozzle, and both pressures are greater than the critical, the complete flow formula (434) must be used. Fluid flow under these conditions is called *affected flow*. For a true gas this may be done, and the results are satisfactorily accurate even though the formula is cumbersome and the computations tedious. But for steam the values of k , C , and R are far from constant. Thus both formulas (434) and (439) usually give erroneous results for steam flow. However, if the value of k is assumed to be 1.138 and the ratio of pressures for maximum flow to be 0.577 or less, equation (433) reduces to

$$(440) \quad M = 3.61 A_0 \sqrt{\frac{P_1}{v_1}}.$$

But, for isothermal change,

$$\frac{P_1}{v_1} = \frac{P_1^2}{P_1 v_1} = \frac{P_1^2}{C},$$

hence (440) becomes

$$(441) \quad M = 3.61 A_0 P_1 \sqrt{\frac{1}{C}}.$$

TABLE 57
VALUES OF C FOR STEAM

P_1	v_1	C	P_1	v_1	C
14.7×144	26.79	56 700	100×144	4.426	63 650
$25. \times 144$	16.30	58 680	200×144	2.285	65 800
40×144	10.49	60 417	400×144	1.160	66 816

For steam the value of C varies as indicated in Table 57, the values of pressure and volume being obtained from tables for dry and saturated steam.

If the value of C for the pressure of 40 pounds per square inch is introduced in (441) it reduces to

$$(442) \quad M = \frac{A_0 P_1}{68.05}.$$

If a coefficient of discharge of 0.972 be allowed (442) reduced to

$$(443) \quad M = \frac{A_0 P_1}{70} = \frac{a_0 p_1}{70},$$

in which a_0 equals the throat area in square inches, and p_1 the absolute pressure in pounds per square inch. Thus there is theoretical justification for Napier's formula. Rankine's derivation of it however was based largely upon empirical methods.¹

188. Grashof's Formula for Steam Flow through Nozzles.²

Let

B = a factor in the Grashof formula,

f = a factor for affected flow,

v_s = specific volume of dry and saturated steam,

v_m = specific volume of 1 lb. of mixture of steam and water,

v_1 = volume of dry steam in a lb. of wet mixture,

v_w = volume of water in a lb. of wet mixture,

x_1 = initial quality of wet steam,

Z = reciprocal of the max. flow value of $\sqrt{r^{2/k} - r^{(k+1)/k}}$.

If instead of $Pv = C$, the relation $Pv^k = K$ is inserted in (434), and the value of k assumed to be 1.064, then for dry and saturated steam at a pressure of 100 lbs. per sq. in. abs.

$$(444) \quad P_1 v_s^k = 144 \times 100 \times 4.429^{1.064} = 70\,150.$$

But for wet steam

$$(445) \quad v_m = x_1 v_s + (1 - x_1) v_w.$$

For values of quality above 50 per cent, the second term of the

¹ See Engineer (London), pp. 352 and 358, Nov. 26 and Dec. 3, 1869.

² See Grashof, *Theoretische Maschinenlehre*.

right-hand member of (445) may be omitted without sensible error. Then by

$$(446) \quad v_s = \frac{v_m}{x_1},$$

(444) may be written

$$(447) \quad P_1 \left(\frac{v_m}{x_1} \right)^{1.064} = K,$$

from which

$$(448) \quad v_m = x_1 \left(\frac{K}{P_1} \right)^{1/1.064} = x_1 \left(\frac{70\ 150}{P_1} \right)^{0.94}.$$

Substituting this value v_m in (440) in place of v_1 , and reducing, one obtains

$$(449) \quad M = \frac{A_0 P_1^{0.97}}{52.47 \sqrt{x_1}}.$$

Since $144\ A_0 = a_0$, and $P_1 = 144\ p_1$, (449) becomes

$$(450) \quad M = \frac{a_0 p_1^{0.97}}{60.9 \sqrt{x_1}} = \frac{0.01642\ a_0 p_1^{0.97}}{\sqrt{x_1}}.$$

If a coefficient of discharge equal to 0.97 is introduced, the result is

$$(451) \quad M = \frac{a_0 p_1^{0.97}}{62.8 \sqrt{x_1}}.$$

Many tests of steam-turbine nozzles made to various specifications indicate that the value of the constant in (451) varies from 61 to 63, depending upon the conditions, hence it may be written

$$(452) \quad M = \frac{a_0 p_1^{0.97}}{B \sqrt{x_1}},$$

in which the value of B varies from 61 to 63. Equations (449) to (452) are known as *Grashof's formulas*. They give results which are seldom in error more than 0.5 of one per cent.

Based upon experimental results, the Grashof formula has been modified for flow of *superheated steam* by A. R. Dodge, of the General Electric Company, as follows:

$$(453) \quad M = \frac{a_0 p_1^{0.97}}{B(1 + 0.00065\ D)}$$

where D equals the number of degrees F. of superheating.

In the affected-flow region where the discharge pressure of the steam leaving the nozzle is greater than $0.58 p_1$, a factor f , less than 1.00, must be introduced. For permanent gases the complete flow formula (432) or (434) can be used. But for steam an

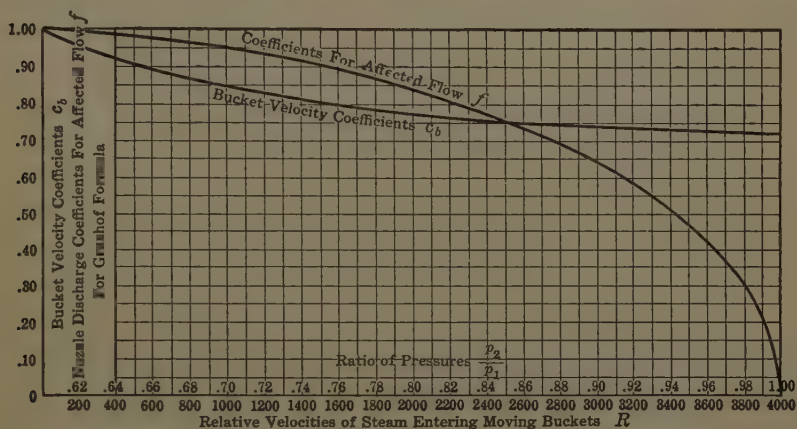


FIG. 177.—BUCKET-VELOCITY COEFFICIENTS AND COEFFICIENTS FOR AFFECTED FLOW

empirical value of f , as follows, gives results which are sufficiently accurate for commercial purposes:

$$(454) \quad f = 0.86 \sqrt{-1 + 8.4r - 7.4r^2}$$

where r is the ratio of final to initial pressure. Or the values of f may be taken from a curve as shown in Fig. 177. Since

$$p^{0.97} = \frac{p}{p^{0.03}},$$

TABLE 58

VALUES OF $p^{0.03}$ AND $p^{0.97}$

p	$p^{0.03}$	$p^{0.97}$	p	$p^{0.03}$	$p^{0.97}$
10	1.017	9.34	160	1.164	137.4
20	1.094	18.28	180	1.167	154.0
40	1.117	35.82	200	1.173	170.6
60	1.131	53.15	250	1.179	212.0
80	1.141	70.15	300	1.187	253.4
100	1.148	87.11	400	1.197	334.2
120	1.154	103.90	600	1.212	495.1
140	1.159	120.70	1200	1.237	970.1

it has been found, at times, convenient to write the Grashof formula

$$(455) \quad M = \frac{a_0 p_1}{B p_1^{0.03} \sqrt{x_1}}.$$

Values of $p^{0.97}$ or $p^{0.03}$ may be readily read from Table 58.

189. Theoretical Efficiency of a Single-Bucket Row.¹ The efficiency of the moving buckets (refer to Fig. 173, and § 184 for symbols) is

$$(456) \quad e_b = \frac{\text{energy supplied} - \text{energy loss}}{\text{energy supplied}}$$

or

$$(457) \quad e_b = \frac{S^2 - S_1^2}{2g} \div \frac{S^2}{2g} = \frac{S^2 - S_1^2}{S^2}.$$

Substituting (405) in (457), and taking $\cos \alpha = 0.94$, one obtains

$$(458) \quad e_b = \frac{4S_b}{S} \left(0.94 - \frac{S_b}{S} \right).$$

The value of S is found from S_0 as follows:

$$(459) \quad S = c_n S_0$$

in which S_0 is found from (423) and c_n , the nozzle-velocity coefficient, has been found by test to vary from 0.95 to 0.98 with 0.96 or 0.97 the more usual value.

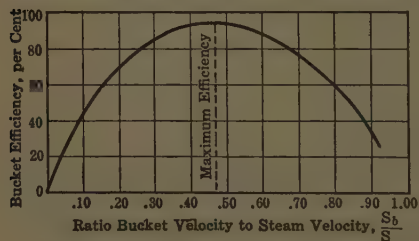


FIG. 178.—THEORETICAL BUCKET EFFICIENCY

the ratio S_b/S approximately as shown in Fig. 178.

When dry and saturated steam is expanded through a properly designed nozzle from an initial pressure of 150 lbs. per sq. in.

¹ See J. A. Moyer, *Steam Turbines*.

absolute down to 2 lbs. per sq. in. absolute, it acquires a velocity S equal to about 3660 feet per second. The bucket velocity necessary to realize maximum efficiency would have to be $3660 \times 0.47 = 1725$ feet per second.

By employing the best materials available in buckets and bucket wheels, the maximum permissible peripheral wheel speed is about 1300 feet per second. Even that is approaching dangerously near to the bursting speed. In American turbines, the bucket velocity S rarely exceeds 900 feet per second. In order to employ lower and safer speeds and still realize high bucket efficiencies, the pressure drop must be broken up into pressure stages and velocity stages as illustrated in Figs. 152, 153, 155, 159, 162, and 200. If high efficiency is to be realized, the steam velocity developed in each stage cannot differ greatly from a value equal to twice the bucket velocity.

Usually the first pressure stage of large impulse turbines, and of most reaction turbines as well, consists of a wheel having mounted thereon two rows of moving buckets with an intermediate row of stationary reversing buckets, in shop language called *intermediates* (see Figs. 149, 153, and 184). This arrangement gives two velocity stages for one pressure stage. The succeeding pressure stages usually have a single velocity stage. In smaller and cheaper machines, there are usually only one or two pressure stages with two velocity stages incorporated in each pressure stage (see Figs. 149, 153, 155, 157, and 162).

190. Actual Efficiency of a Double-Bucket Row. The diagrams illustrated in Fig. 180 constitute a single pressure stage with two velocity stages. In these as well as all other velocity diagrams, the symbol S with proper subscript refers to the respective absolute steam velocities; R with proper subscript refers to the respective relative steam velocity, and S_b always denotes the velocity of the buckets.

In order to solve the series of velocity diagrams, R_1 must be obtained from R , and R_3 from R_2 . Theoretically R_1 should equal R , and R_3 should equal R_2 . However there are velocity losses due to eddies, surface friction, and shock as the steam enters and

passes through both the stationary and the moving buckets. The extent of these losses has been found by experiment to vary with the value of the *relative velocity* of the steam entering the moving buckets. It is practically constant for the stationary buckets. The ratio of the exit velocity to the inlet velocity



FIG. 179.—PRESSURE AND VELOCITY HISTORY FOR DOUBLE BUCKET ROW

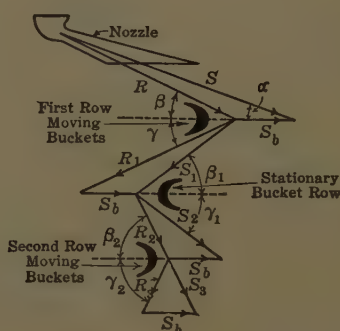


FIG. 180.—DOUBLE-BUCKET ROW

for the moving buckets depends primarily upon the magnitude of relative velocity of the entering steam. It is most conveniently expressed in the form of a bucket-velocity coefficient as follows:

$$(460) \quad c_b = \frac{R_1}{R} = \frac{R_3}{R_2}.$$

The values of c_b have been determined experimentally from a large number of tests. The values as shown in Table 59 or by the curve of Fig. 177 may be used for any row of moving buckets.

The symbol R in the table indicates the *relative inlet velocity* of the steam for the row of moving buckets under consideration.

TABLE 59
BUCKET-VELOCITY COEFFICIENTS¹

R	200	400	600	800	1000	1500	2000	3000	4000
c_b	0.953	0.918	0.888	0.863	0.841	0.801	0.774	0.737	0.716

For the stationary buckets of the intermediates, the velocity coefficient is constant at about 0.95 for all types and sizes of buckets.

The actual velocity S of the steam leaving the nozzle and entering the first row of moving buckets is somewhat less than the theoretical spouting velocity S_0 as already explained (see equation 424).

By referring to Fig. 180 and equation (457), it will be readily seen that the expression for the nozzle-and-bucket efficiency e_b for a double-bucket row is

$$(461) \quad e_b = \frac{S_0^2 - (S_0^2 - S^2) - (R^2 - R_1^2) - (S_1^2 - S_2^2) - (R_2^2 - R_3^2) - S_3^2}{S_0^2}.$$

But by assuming c_n of (424) equal to 0.97 and solving, the result is

$$(462) \quad S_0^2 = \frac{S^2}{0.94}.$$

Substituting (462) in (461), combining and simplifying, one obtains

$$(463) \quad e_b = \frac{0.94 (S^2 - S_1^2 + S_2^2 - S_3^2 - R^2 + R_1^2 - R_2^2 - R_3^2)}{S^2}.$$

The various values of S and R in (463) must be obtained by use of the cosine formula (404), and the value of S_0 by use of equation (423). A problem worked out step by step will best illustrate the necessary procedure.

¹ See Marks' *Mechanical Engineers' Handbook*, p. 1036.

191. Nozzle-and-Bucket-Efficiency Problem. Given an impulse steam turbine which operates under the following conditions: The turbine has a single pressure stage provided with two velocity stages as shown in Fig. 149.

Initial pressure $p_1 = 125$ lbs. per sq. in. abs.

Final pressure $p_2 = 15$ lbs. per sq. in. abs.

Initial quality $x_1 = 1.00$

Nozzle-velocity coefficient $= 0.96$

Nozzle exit angle $= 20^\circ$

Exit angle 1st row of moving buckets $= 24^\circ$

Exit angle of intermediates $= 26^\circ$

Exit angle of 2d row of moving buckets $= 35^\circ$

Pitch diameter of wheel $= 36''$

R.p.m. of wheel $= 3600$.

To simplify the problem, the entrance and exit angles of all buckets are assumed to be equal.

Computations:

From tables cosine $20^\circ = 0.940$

cosine $24^\circ = 0.914$

cosine $26^\circ = 0.899$

cosine $35^\circ = 0.819$

Bucket velocity $S_b = \frac{3.1416 \times 3 \times 3600}{60} = 565$ ft. per sec.

$$S_b^2 = 319\,225.$$

From (423) and the Mollier diagram

$$S_0 = 223.8 \sqrt{1191. - 1035} = 2794 \text{ ft. per sec.}$$

$$S = 0.96 S_0 = 2794 \times 0.96 = 2680 \text{ ft. per sec.}$$

By use of cosine formula (404)

$$R = \sqrt{S^2 + S_b^2 - 2 \cos 20^\circ S S_b}.$$

Attention is especially directed to the fact that the solution of the cosine formula involves the sum and the difference of some large numbers. *Hence the results obtained with a 10-inch slide rule are not sufficiently accurate.* In fact, such slide-rule results may be absurd under these conditions.

$$R = \sqrt{2680^2 + 565^2 - 2 \times 0.94 \times 2680 \times 565} = 2158 \text{ ft. per sec.}$$

From Table 59, or curve in Fig. 177, $c_b = 0.767$, hence

$$R_1 = 2158 \times 0.767 = 1654 \text{ ft. per sec.}$$

Again, by cosine formula,

$$S_1 = \sqrt{1654^2 + 565^2 - 2 \times 0.914 \times 1654 \times 565} = 1160 \text{ ft. per sec.}$$

As explained on page 421 the velocity loss through stationary buckets is a constant percentage regardless of the entering velocity, hence

$$S_2 = 0.95 S_1 = 1160 \times 0.95 = 1102 \text{ ft. per sec.}$$

Again, by the cosine formula,

$$R_2 = \sqrt{1102^2 + 565^2 - 2 \times 0.899 \times 1102 \times 565} = 645 \text{ ft. per sec.,}$$

$$R_3 = 645 \times 0.882 = 568 \text{ ft. per sec.,}$$

and

$$S_3 = \sqrt{568^2 + 565^2 - 2 \times 0.819 \times 568 \times 565} = 341 \text{ ft. per sec.}$$

Collecting these values as indicated in (463), and using a nozzle-velocity coefficient of 0.96, one finds that the nozzle-and-bucket efficiency is

$$e_b = \frac{0.922(\overline{2680^2} - \overline{1160^2} + \overline{1102^2} - \overline{341^2} - \overline{2158^2} + \overline{1654^2} - \overline{645^2} + \overline{568^2})}{\overline{3680^2}} = 0.632$$

The various velocities in the above equation represent definite quantities of energy which may be expressed in foot-pounds or in B.t.u. per lb. of steam per second. It may give a little clearer exposition of the disposition of the heat as the steam passes through the nozzle and buckets to convert the energies represented by the various velocities into B.t.u. This may be done by use of (423), as follows:

$$E = H - H = \frac{S^2}{223.8^2}.$$

Then the energies may be tabulated either as velocities or heat units. The velocities are arranged in the order shown in (461).

SYMBOL	STEAM VELOCITIES FT. PER SEC.	ENERGY IN B.T.U. PER SEC.	SYMBOL	STEAM VELOCITIES FT. PER SEC.	ENERGY IN B.T.U. PER SEC.
$S_0 =$	2794	156.0	$S_2 =$	1102	24.3
$S =$	2680	144.0	$R_2 =$	645	8.3
$R =$	2158	93.2	$R_3 =$	568	6.4
$R_1 =$	1654	54.7	$S_3 =$	341	2.3
$S_1 =$	1160	26.9			

The above symbols refer to velocities as shown in Fig. 180.

The losses per lb. of steam per second may now be determined in B.t.u. and retabulated as follows:

Nozzle loss	=	156.0 - 144.0 = 12.0 B.t.u.
First moving-bucket loss	=	93.2 - 54.7 = 38.5 "
Stationary-bucket loss	=	26.9 - 24.3 = 2.6 "
Second moving-bucket loss	=	8.3 - 6.4 = 1.9 "
Residual velocity loss	=	2.3 "
Total loss		= 57.3 "

And the efficiency of nozzle and buckets may be expressed in terms of B.t.u. thus:

$$e_b = \frac{\text{energy supplied} - \text{energy loss}}{\text{energy supplied}} = \frac{156 - 57.3}{156} = 0.633$$

The values of e_b as obtained from the velocities and from the B.t.u. should check exactly. This would be the case if all the numerical work involved were executed with the required degree of accuracy.

192. Rotation Losses in Impulse Turbines. The nozzle-and-bucket losses as above determined do not constitute all of the turbine losses. There are losses due to the windage and friction of the wheel discs and the buckets rotating at high speeds in the atmosphere of steam surrounding them. Under certain conditions the rotation losses constitute a large part of the entire energy losses. In addition there is frictional loss in the bearings of the turbine.

Many formulas have been proposed for computing rotation losses. On account of the large number of variables which bear on the problem, the results obtained by the use of rationally developed formulas often give very inaccurate results. It is difficult to segregate and measure rotation losses experimentally.

The usual method employed to measure rotation losses is to operate the turbine under load and measure the maximum horsepower developed as the number of nozzles open to steam flow is varied from the maximum down to the smallest number which are required to turn the turbine wheels at normal speed without useful load. Curves are then plotted between horsepower or brake force and the number of nozzles open. The resulting curve is usually a

straight line, as shown by curve A, Fig. 181. The distance a on the vertical axis measured in horsepower represents the power absorbed in all the rotation losses and the bearing losses combined.

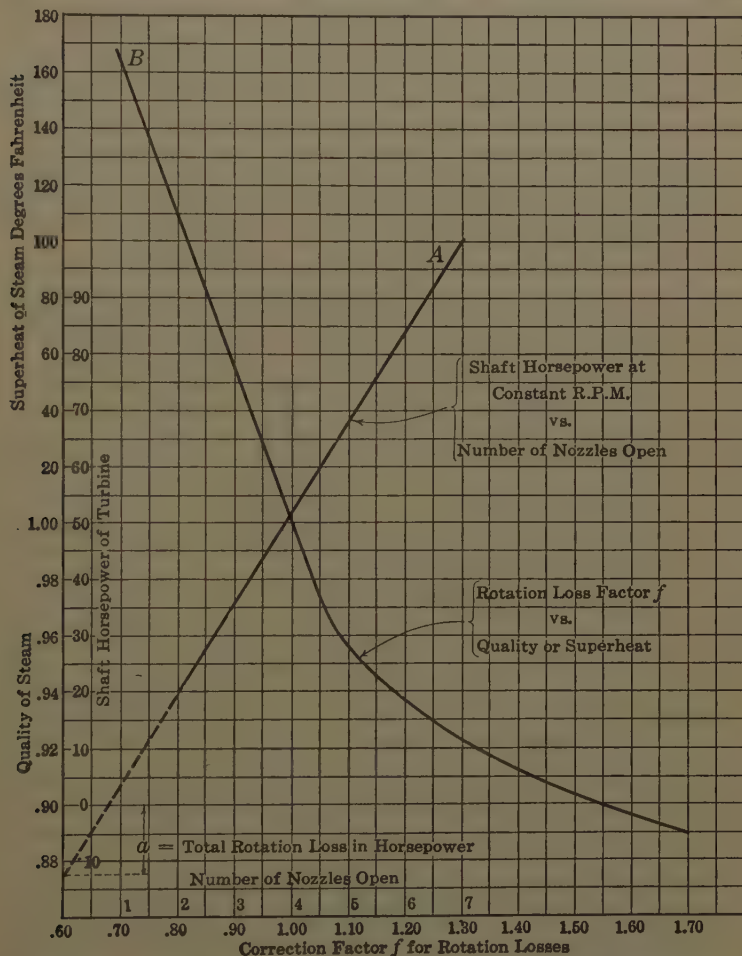


FIG. 181.—TOTAL ROTATION LOSS AND ROTATION-LOSS CORRECTION

Rotation losses may be separated into two parts. One is due to the skin friction of the wheel disc alone. The other is due to the buckets mounted on the rim of the disc cutting through the surrounding steam. The entire surface of the disc is active in producing loss, but only those buckets which are not covered by

spouting nozzles are active in producing frictional loss. Theory indicates that the loss should vary approximately as the cube of the speed. Tests show that the value of the exponent varies from 2.5 to 3.5, depending upon conditions. In small turbines, the total rotation loss at full load may amount to as much as 40 per cent of the energy supplied to the turbine. In machines having capacities varying from 10 000 to 100 000 kilowatts, the loss usually varies from 5 to 15 per cent of the energy supplied.

The following formulas have been developed, partly from theory and partly by empirical methods, and have been found to apply rather closely to the conditions prevailing in the Curtis and Rateau types of impulse turbines. In addition to the symbols already defined, the following will be used.

b = nozzle-and-bucket loss (a decimal fraction of the total energy supplied),

D = pitch diameter of the bucket wheel, ft. measured from center line to center line of buckets,

δ = density of dry and saturated steam at the pressure surrounding the wheel,

e_w = efficiency of turbine wheel,

E = the energy in B.t.u. per lb. of steam available from the pressure drop in the stage under consideration,

f = a factor by which y is to be corrected when the steam is wet or superheated (f is read from Fig. 181),

h = mean height of moving buckets in feet, (if there are two rows of moving buckets— $h = (h_1 + h_2)/2$),

Ke = kinetic energy in ft.-lbs.,

kw = kilowatts,

M = lbs. of steam flowing per sec.,

N = revolutions per minute,

n = number of rows of moving buckets on the wheel,

q = fraction of the wheel circumference not covered by spouting nozzles,

r_b = rotation loss in kw. due to buckets,

r_w = rotation loss in kw. due to wheel disc,

y = total rotation loss as a decimal fraction of E .

Experiment has shown that the rotation losses may be represented with commercial accuracy by the following expressions:

$$(464) \quad r_b = 4 \left(\frac{S_b}{100} \right)^3 nhq\delta D,$$

and

$$(465) \quad r_w = 0.19 \left(\frac{S_b}{100} \right)^3 \delta D^2.$$

The total rotation loss in kilowatts equals the sum of (464) and (465):

$$(466) \quad r_b + r_w = 0.19 \left(\frac{S_b}{100} \right)^3 \delta D^2 \left[1 + 21nq \frac{h}{D} \right].$$

The available kinetic energy in kilowatts per second delivered from the nozzle in the spouting steam is

$$(467) \quad kw = \frac{MS_0^2}{64.34 \times 550 \times 1.341} = \frac{MS_0^2}{47\,460}.$$

Therefore the rotation losses as a decimal fraction of the available energy equals

$$(468) \quad y = \frac{0.19 \left(\frac{S_b}{100} \right)^3 \delta D^2 \left[1 + 21nq \frac{h}{D} \right]}{\frac{MS_0^2}{47\,460}}.$$

By (423)

$$S_0^2 = 50\,086 E.$$

Hence introducing this value of S_0^2 into (468) and reducing, one obtains

$$(469) \quad y = \frac{0.1804 \left(\frac{S_b}{100} \right)^3 \delta D^2 \left[1 + 21nq \frac{h}{D} \right]}{ME}.$$

Since

$$S_b = \frac{3.1416DN}{60},$$

(469) may be reduced to

$$(470) \quad y = \frac{\delta D^5}{ME} \left[1 + 21nq \frac{h}{D} \right] \left[\frac{N}{3380} \right]^3.$$

Equation (470) is correct only when the atmosphere of steam in which the wheel rotates is just dry and saturated. For wet steam or for superheated steam a factor read from curve *B*, Fig. 181, must be introduced. A study of this curve shows that there is a distinct advantage to be gained by supplying superheated steam to the turbine. For wet steam or superheated steam, (470) becomes

$$(471) \quad y = \frac{f \delta D^5 \left(1 + 21nq \frac{h}{D} \right)}{ME} \left(\frac{N}{3380} \right)^3.$$

Then

$$(472) \quad e_w = 1.00 - (b + y).$$

In designing the older turbines it was general practice, and is yet for the smaller sizes, to employ only a few nozzles which extend over only a small arc of the wheel in the early stages. In recent designs of large capacity it is customary to place nozzles nearly or entirely around the wheels in all stages. In this way the value of *n* in (471) is made nearly equal to zero, thus materially reducing the rotation loss for the entire turbine. In a large measure this accounts for the appearance of the rotors in high-efficiency machines (see Fig. 199).

The energy loss represented by (*b* + *y*) due to friction, eddies, and shock is not a complete loss provided there is a subsequent stage or stages in which to utilize it. This energy appears in the form of heat which goes to raise the quality of the steam or to superheat it, as the case may be, before it passes from the stage under consideration. When the steam passes into the succeeding stage it carries with it in the form of available heat all the losses of the stage except one or two per cent which is lost beyond recovery by radiation from the turbine. Herein lies one of the important advantages of the multistage turbine. By adding stages the ultimate losses by windage and friction may be reduced to a small value. The economic limit is reached when the sum of the increased cost, the total leakage of steam between stages, and the increased bearing-friction loss equals the gain due to additional stages. In large impulse turbines from 15 to 30 pressure stages may be employed profitably. In the large reaction turbines 40 to 60 stages are being used.

193. The Path of the Steam through the Turbine on the Mollier and TN -Planes. If, in an impulse turbine, the entire pressure drop occurred in a single stage, the heat made available in a Rankine cycle would be shown by the straight vertical line ab' in Fig. 182. But if it occurred in several stages, the pressure drop in the first-stage nozzles would be represented by the line ab . The losses caused by the first-stage nozzles, buckets, and wheel would raise the heat content of the steam at constant pressure along the line bc . The second-stage pressure drop would occur adiabatically along the line cd with subsequent reheating,

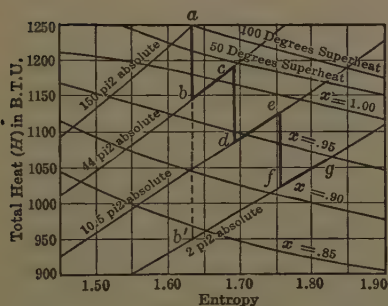


FIG. 182.—PATH OF THE STEAM THROUGH A TURBINE

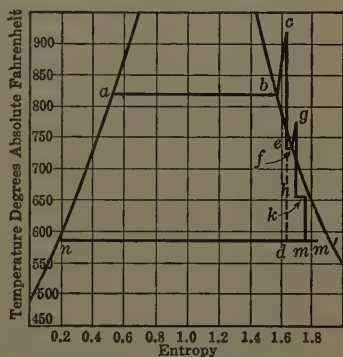


FIG. 183.—STEAM PATH ON THE TN PLANE

at constant pressure along the line de , due to the losses and likewise throughout all the succeeding stages of the turbine. The zig-zag line $abcdefg$ is known as the *path of the steam through the turbine*. The path of the steam on the TN -plane is shown in Fig. 183. The paths in Figs. 182 and 183 are for a three-stage turbine. In general there would be as many vertical or constant entropy steps as these are pressure stages.

When steam expansion is broken up into several stages with recovery of the reheat, the quality of the steam as it passes through the turbine remains considerably higher with correspondingly smaller rotation losses. The sum of the rotation losses in all the stages of a turbine amounts to from 10 to 40 per cent of the total available energy, most of which is recovered. This recovered energy is known as *reheat energy*. In designing large tur-

bines, the reheat factor for each stage is determined from a table compiled from the accumulated experience of turbine designing and testing. In designing small turbines below 5000 horsepower capacity, the equations developed on pages 426 to 428 inclusive may be employed in computing the rotation losses.

Hence it is seen that the available energy in each stage of a turbine is the sum of the energy due to the stage pressure drop plus from 10 to 40 per cent of the energy due to the pressure drop of the preceeding stage. The percentage of energy carried over in each case depends upon the magnitude of the rotation losses.

For the conditions of steam pressure and superheat shown in Fig. 183, the single-stage Rankine-cycle area is $abcdn$. By breaking the pressure drop up into three stages, there is a gain in the total available energy due to the reheat energy as indicated by the area $abceghkm$. However, only a portion of this extra energy made available by reheat can be actually converted into useful work at the turbine shaft because the efficiency of utilization of any portion of the energy supplied can never equal 100 per cent. In three-stage turbines the nozzle-and-bucket efficiency may vary from 15 to 60 per cent. The chief gain from the introduction of additional stages, if it is not overdone, is due to the fact that the ratio of bucket to steam velocity approaches nearer the ideal of 0.47. Hence the area $efghkmd$ represents only a part of the increased wheel efficiency due to the introduction of a number of pressure stages.

194. Steam Leakage between Stages. The stages of an impulse turbine are separated by diaphragms. Some form of packing gland around the shaft is always provided in the diaphragms and the turbine cylinder heads to prevent as far as possible the leakage of steam from one stage to the next one, in which the pressure is lower, and to prevent leakage into the atmosphere. The packing cannot fit tightly upon the shaft as is the case with piston rods of engines because the mechanical friction loss due to the high rotative speeds of turbine shafts would be excessive. This problem is met by making the packings in sectionalized rings which are about 0.02 inch larger in diameter

than the shaft. Different forms of packings are shown in Figs. 155 and 159. The packing rings, made of metal alloys or carbon, are mounted in housings in the diaphragm which hold them concentric with the shaft. Thus there is an annular ring about 0.01 inch thick through which a small amount of steam leaks from stage to stage and into the atmosphere. The leakage amounts to one or two per cent of the steam flow in each stage.

In computing the leakage loss it is generally assumed that the velocity of steam flow through the small annular orifice is equal to the theoretical spouting velocity due to the pressure difference on the two sides of the packing. The area of the steam jet through the packing is considered to be equal to the full area of the annular ring between shaft and packing. The actual jet area is probably not more than 70 or 80 per cent of this because the *vena contracta* effects are considerable. But the total leakage is so small, and it is always on the side of conservatism as to the computed steam rate to neglect this correction. Hence m , the pounds of leakage steam per second, as computed from the continuity equation equals

$$(473) \quad m = \frac{AS}{v} = \frac{0.01 \times 3.1416 \times 223.8d\sqrt{E}}{144v} = \frac{0.049d\sqrt{E}}{v}.$$

For high-pressure turbines (600 and 1200 lbs.) elaborate labyrinth packings are provided at the end of the turbine cylinder and between the stages at the high-pressure end of the turbine (see Fig. 204).

195. Bearing Loss. The bearing loss due to friction seldom amounts to more than one or two per cent of the turbine horsepower. This very small loss is due to the fact that turbine shafts run at high rotative speeds and the bearings are supplied with oil under a pressure varying from 5 to 50 pounds per square inch; thus there is perfect lubrication and the loss is only that due to the internal friction of the lubricant.

196. Selection of Steam Pressures for a Three-Stage Turbine. Usually, in small turbines having only three or four stages, the turbine wheels are all made equal in diameter, and the most advantageous

result is obtained when the energy drops are nearly equal in all stages. The selection of the pressures to give this result is largely a matter of trial-and-error, and experience. A concrete example is given to illustrate the method.

p_1 = initial steam pressure, lbs. per sq. in. abs.,

p_2 = 1st-stage shell pressure and the pressure of the steam entering the 2d-stage nozzles, lbs. per sq. in. abs.,

p_3 = the 2d-stage shell pressure and the pressure of the steam entering the 3d-stage, lbs. per sq. in. abs.,

p_4 = the exhaust pressure, lbs. per sq. in. abs.

p_1 given as 150 lbs. per sq. in. abs.

p_4 given as 2 lbs. per sq. in. abs.

Initial superheat 100 degrees F.

Reheat factor assumed to be 0.40.

The Mollier diagram will be used.

First, guess the pressures to be $p_2 = 50$; and $p_3 = 15$ lbs. per sq. in. abs. Then for the first stage and one pound of steam flowing per second

$$H_1 = 1251 \text{ B.t.u.}$$

$$H_2 = 1156 \text{ B.t.u.}$$

$$H_1 - H_2 = E_1 = 95 \text{ B.t.u.}$$

For the second stage

$${}_2H_1 = 1156 + (95 \times .4) = 1194 \text{ B.t.u.}$$

$${}_2H_2 = (\text{from Mollier diagram}) 1103 \text{ B.t.u.}$$

$${}_2H_1 - {}_2H_2 = E_2 = 91 \text{ B.t.u.}$$

For the third stage

$${}_3H_1 = 1103 + (91 \times .4) = 1139 \text{ B.t.u.}$$

$${}_3H_2 = \text{from Mollier diagram} = 1009 \text{ B.t.u.}$$

$${}_3H_1 - {}_3H_2 = E_3 = 130 \text{ B.t.u.}$$

and

$$E_1 + E_2 + E_3 = 95 + 91 + 130 = 316 \text{ B.t.u.}$$

For a second trial assume $p_2 = 44$; and $p_3 = 10.5$ lbs. per sq. in. abs. As before, the reheat factor is taken as 0.40 for each stage. Then for the first stage,

$$H_1 = 1151 \text{ B.t.u.}$$

$$H_2 (\text{from Mollier diagram}) = 1146 \text{ B.t.u.}$$

$$H_1 - H_2 = E_1 = 105 \text{ B.t.u.}$$

For the second stage

$${}_2H_1 = 1146 + (105 \times .4) = 1188 \text{ B.t.u.}$$

$${}_2H_2 \text{ (from Mollier diagram)} = \underline{1183 \text{ B.t.u.}}$$

$${}_2H_1 - {}_2H_2 = E_2 = 105 \text{ B.t.u.}$$

For the third stage

$${}_3H_1 = 1183 + (105 \times .4) = 1125 \text{ B.t.u.}$$

$${}_3H_2 \text{ (from Mollier diagram)} = \underline{1020 \text{ B.t.u.}}$$

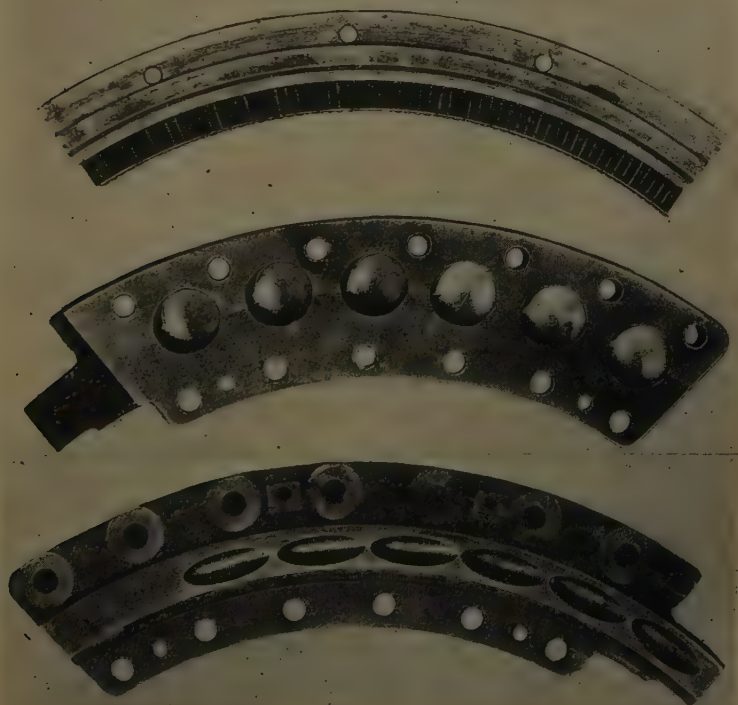
$${}_3H_1 - {}_3H_2 = E_3 = 105 \text{ B.t.u.}$$

and

$$E_1 + E_2 + E_3 = 105 + 105 + 105 = 315 \text{ B.t.u.}$$

Hence it is seen that the first series of pressures are slightly better as regards total available energy.

For multistage turbines, the process of selecting the most advantageous stage pressures is a step-by-step process somewhat more involved.



(General Electric Co.)

FIG. 184.—NOZZLE PLATE AND INTERMEDIATES FOR CURTIS TURBINE

197. Nozzles. Nozzles of circular section are nested together in the earlier stages of a turbine in annular plates as illustrated in Fig. 184. In the first stages of small turbines, they do not usually cover more than $1/6$ or $1/3$ of the wheel circumference. As the steam progresses from stage to stage, its specific volume increases; hence the number of nozzles, the nozzle-throat areas, and the height of the buckets all have to increase progressively to care for the increased steam volume. In the later stages of large turbines, they may be rectangular in cross-section, and are formed by casting sheet-metal vanes, properly curved, into the diaphragms to give the correct nozzle outline. The limit to the number of nozzles is reached when they completely fill the circumference of the wheel. Hence there are several factors which enter this nozzle problem. These factors are susceptible of considerable adjustment which can best be worked out by trial-and-error methods. The combinations of factors employed by different impulse-turbine manufacturers is usually the result of considerable experience and represents the best that their engineers have been able to arrange consistent with minimum or reasonable costs of the various classes and sizes of turbines.

198. Methods of Computing Turbine Horsepower and Steam Rates. The steps outlined below, in their proper sequence, for a two-stage turbine, modified as required for different numbers of stages, are employed in computing the power delivered by and steam consumptions of small turbines ranging in full-load capacity from one horsepower to perhaps 3500 or 4000 horsepower. The number of stages may vary from one to four. The symbols used are explained below. All pressures are understood to be in lbs. per sq. in. abs., and all energies taken from the Mollier chart are B.t.u. per lb. of steam.

c_n = nozzle-velocity coefficient, usually constant for all stages, the value varies from 0.95 to 0.98 depending upon the type and size of nozzles,

δ_2 = density of dry and saturated steam at pressure p_2 ,

δ_3 = density of dry and saturated steam at pressure p_3 ,

- E_1 = B.t.u. made available by pressure drop through the first-stage nozzles, by a flow of 1 lb. of steam per second,
 E_2 = B.t.u. made available by pressure drop through the second-stage nozzles when 1 lb. of steam flows per sec.
 e_1 = nozzle-and-bucket efficiency in the first stage,
 e_2 = nozzle-and-bucket efficiency in the second stage,
 e_R = Rankine-cycle efficiency,
 e_r = efficiency ratio,
 e_t = thermal efficiency for the complete turbine,
 $1e_w$ = efficiency of nozzle, buckets, and wheel after deducting rotation loss for the first stage,
 $2e_w$ = efficiency of nozzle, buckets, and wheel after deducting rotation-loss for the second stage.
 f = rotation-loss correction factor for wet or superheated steam,
 H_1 = B.t.u. above 32° F. for 1 lb. of steam at pressure p_1 ,
 H_2 = B.t.u. above 32° F. for 1 lb. of steam, after adiabatic expansion from p_1 to p_2 , read from Mollier diagram,
 H_3' = estimated B.t.u. in 1 lb. of steam after adding the estimated reheat. It is used to find the quality in computing rotation loss in the first stage,
 H_3 = B.t.u. per lb. of steam entering the second-stage nozzles,
 H_4 = B.t.u. per lb. of steam after adiabatic expansion from p_2 to p_3 in the second-stage nozzles,
 H_4' = estimated B.t.u. per lb. of steam after adding the estimated reheat. It is used to find the quality in computing second-stage rotation loss,
 h_3 = heat of the liquid in B.t.u. per lb. at pressure p_3 ,
 M_1 = lbs. of steam per sec. flowing through the first-stage nozzles,
 M_2 = lbs. of steam per sec. from second-stage nozzles,
 p_1 = steam pressure entering the first-stage nozzles,
 p_2 = pressure in the first-stage shell. This is the terminal pressure for the first-stage nozzles and the initial pressure for the second-stage nozzles,

p_3 = pressure in the second-stage shell. It is slightly above the exhaust pipe pressure, perhaps a half-pound or so,

Q_1 = B.t.u. per lb. of steam available for work upon the first-stage wheel,

Q_2 = B.t.u. per lb. of steam available for work upon the second-stage wheel,

q_1 = fraction of wheel circumference not covered by spouting nozzles in the first stage,

q_2 = fraction of wheel circumference not covered by spouting nozzles in the second stage,

r = the reheat factor; it varies from 0.10 to 0.40,

FP = friction horsepower of turbine bearings,

SP = shaft horsepower available for useful work outside the turbine,

WP = wheel horsepower of the turbine,

SR = turbine steam rate, lbs. per shaft horsepower per hour.

Then for the first stage

$$(474) \quad E_1 = H_1 - H_2,$$

$$(475) \quad S_0 = 223.8 \sqrt{E_1},$$

$$(476) \quad S = c_n S_0.$$

The first-stage nozzle-and-bucket efficiency e_1 is found by solving the velocity diagrams, Fig. 180, as illustrated in § 191.

$$(477) \quad H_3' = H_2 + r E_1.$$

The value of the reheat factor should be chosen with some care. It varies with the size of the turbine, the number of stages, and with the initial and final steam pressures. Good judgment in selecting the value can only be acquired by experience. The value can be selected by the trial-and-error method also. A guess as to the value of r is made, from the value of H_3' thus obtained the quality of the steam at pressure p_2 is used in computing the first-stage rotation loss. If the nozzle, bucket, and rotation losses give a computed reheat much different from the estimated

value of rE_1 then the information thus gained is used in selecting a new value of r more nearly correct. Usually a second trial is sufficiently accurate.

$$(478) \quad y_1 = \frac{f_1 \delta_2 D^5 \left[1 + 21 n q \frac{h}{D} \right]}{M_1 E_1} \left[\frac{N}{3380} \right]^3.$$

$$(479) \quad {}_1e_w = e_1 - y_1,$$

$$(480) \quad Q_1 = E_1({}_1e_w),$$

$$(481) \quad H_3 = H_2 + (E_1 - Q_1),$$

$$(482) \quad E_2 = H_3 - H_4.$$

Then the series of steps as already outlined for the first stage are repeated for the second stage to find Q_2 .

$$(483) \quad M_2 = M_1 - (\text{the steam leakage between stages}).$$

Then

$$(484) \quad \begin{aligned} WP &= \frac{3600 (M_1 Q_1 + M_2 Q_2)}{2543} \\ &= 1.416 (M_1 Q_1 + M_2 Q_2). \end{aligned}$$

At full load the bearing friction usually equals about one per cent of the wheel horsepower, and it is nearly constant for all loads on the turbine. Hence

$$(485) \quad SP = 0.99 WP,$$

and

$$(486) \quad SR = \frac{3600 M_1}{SP}.$$

To obtain a steam-rate curve for various horsepowers, values of shaft horsepower should be computed for at least two rates of steam flow by the method outlined above. The flows at full-rated load, and at about one-half or one-third load may be conveniently chosen. From these two computed points a Willans line may be drawn from which the steam rates at all fractional loads may be readily determined. Steam rates computed in this manner may be in error from 1 to 5 or 6 per cent for values below

one-third load and for overloads. But such results are sufficiently accurate for most purposes. If more accurate results are desired, it is necessary to compute steam rates as outlined by equations (474) to (486) for each fractional load. The reference given below¹ contains reports of several careful steam-rate tests of General Electric Curtis-type turbines, and Westinghouse Parsons-type turbines with the accompanying Willans lines. These tests were carried out on modern high-pressure turbines varying from 6 000 to 60 000 kw. rated capacity. In nearly every case for which test results are given the Willans line is a straight line between about 25 per cent rated load and full load.

For small one-stage, two-stage, and three-stage impulse turbines, the Willans line holds remarkably well even down to 10 per cent of the rated load. But at extremely small loads and at overloads for all sizes of turbine, the steam flow as determined by test is from 1 to 5 per cent greater than indicated by the straight line. Thus the correct Willans line is made up of a series of broken straight lines or a curve convex to the horsepower axis.

199. Turbine Efficiencies. It is not possible to determine the indicated and friction horsepower of any form of steam turbine as is the case for steam engines. Hence there are only four efficiencies which may be computed for turbines in place of six for steam engines. The ideal Rankine cycle serves as a standard of perfection with which to compare turbine performance. The total over all thermal efficiency referred to the shaft horsepower or the thermal efficiency based upon the useful-power output of a turbo-generator² measured at the switchboard is the measure of turbine performance universally used in comparing turbines. The efficiency ratio completes the list of turbine efficiencies generally computed. Since the arrival of high-pressure regenerative, and reheat cycles, turbine performance is generally expressed in terms of the number of B.t.u. supplied to it in the steam per horsepower-hour or per kilowatt-hour of output.

As in the case of the steam engine, the efficiency of the cor-

¹ See Proceedings National Electric Light Assn., 1926, 1927, and 1928.

² See Power Plant Engineering, April 15, 1930, p. 430, for use of the words *turbo-generator* and *turbine-generator*.

responding ideal Rankine cycle is

$$(487) \quad e_R = \frac{H_1 - H_2}{H_1 - h_2}$$

in which H_2 and h_2 are respectively the total heat in a pound of steam after adiabatic expansion from the initial pressure to the exhaust pressure, and the heat of the liquid at the exhaust pressure. The thermal efficiency is

$$(488) \quad e_t = \frac{2543}{SR(H_1 - h_2)},$$

and the Rankine efficiency ratio is

$$(489) \quad e_r = \frac{e_t}{e_R} = \frac{2543}{SR(H_1 - H_2)}.$$

200. Reaction Turbines. Reaction-turbine rotors are made in the form of a cylindrical drum. They are usually made up in two or three sections which progressively increase in diameter toward the exhaust end of the turbine. In order to care for the increasing volume of the expanding steam, the successive rows of blades are made longer and wider. These features are illustrated in Figs. 193 to 196 inclusive. In the largest machines the blades may be from 24 to 34 inches long in the last row. The casings for reaction blading are usually bored conically throughout so that the radial clearances between the tips of the blades and the casing may be made as small as possible. Small radial clearances are an important feature in reaction turbines since there is considerable pressure drop across each row of blades which may produce serious leakage of steam between the stages. This requirement is the cause of much difficulty in designing the rotors sufficiently rigid and with a minimum of expansion and contraction effects due to temperature changes.

The entrance and exit angles of reaction blades are unequal; hence large axial thrusts are produced on the rotor. In all of the smaller machines and many of the larger ones, Kingsbury thrust bearings are used to care for this unbalanced axial force. The location of the thrust bearing is shown in Figs. 192 and 193; but

in the largest machines the thrust is too great to be taken care of by thrust bearings alone. In these machines flow of steam in both directions axially along the drum is provided for, and most of the end thrust is thus balanced out. This type of construction is illustrated in Figs. 195 and 196. However there is usually a small residual of unbalance which may change its direction with different loads on the turbine. This residual of unbalance is taken care of by a thrust bearing of moderate proportions.

In impulse turbines, it is general practice to call the vanes *buckets*, and in reaction turbines to call them *blades*. In an impulse turbine, one stage consists of a set of nozzles and one or more rows of moving buckets mounted on one wheel together with the necessary stationary buckets. In this country, one stage in a reaction turbine is considered to consist of a row of stationary blades and its companion row of moving blades. In European countries, a reaction-turbine stage is considered to consist of a single row of blades either moving or stationary.

201. Reaction Turbine Velocity Diagrams. The arrangement of blades and the corresponding velocity diagrams for one stage of a reaction turbine are shown in Figs. 185, and 186. The blades in the two rows are generally exactly alike in form. But the

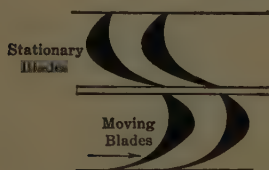


FIG. 185.—REACTION
BLADES

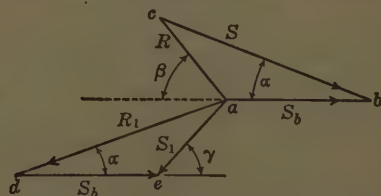


FIG. 186.—REACTION VELOCITY
DIAGRAM

direction of the curvature of the surfaces is reversed. The inlet and exit angles in the two rows are respectively equal. Also in the velocity diagram, angle β equals angle γ . The exit angles α are generally made 20 degrees as in the case of impulse nozzles. Sometimes when the design formula indicates that in order to properly expand the steam, an odd number of rows or a fractional row of blading is required, the nearest even number of rows are

used, as for example, the formula might indicate 75.4 rows of blades or 75.8 rows; in this case the turbine would be constructed with 76 rows. Then the exit angles of the final two or three rows are changed slightly to give the proper area to accommodate the steam flow by twisting the blades with a key after they have been attached to the turbine rotor.

Referring to the velocity diagram, Fig. 186, it is seen that the steam leaves the stationary blades with the absolute velocity S , enters the moving blades with the relative velocity R , and leaves with the relative velocity R_1 and with the absolute velocity S_1 . The absolute velocity of the moving blades is S_b . The relative magnitudes and directions of these various velocities are shown in the vector triangles. In impulse turbines, R_1 is always slightly less than R , but in reaction turbines the velocity is augmented as the steam passes through both stationary and moving blades due to the progressive pressure drop which occurs in the rows of blades and not in nozzles as in impulse turbines. The realization of high efficiency requires that R_1 and S shall be equal. Then the two triangles are equal. The corresponding parts which are also equal being S , R_1 ; S_1 , R ; and S_b is common to both. In order that the corresponding parts of the triangles shall be equal, the energy made available by expansion must be equal in the stationary and moving rows of blades. Theoretically this may be so if the blades in the two rows are symmetrical in every respect. Then, neglecting friction, one has the following relation ¹

$$(490) \quad \frac{R_1^2}{2g} = \frac{S^2}{2g} = \frac{S_1^2}{2g} + JE,$$

in which E represents the B.t.u. made available by adiabatic pressure drop across a single row of blades (one-half a stage). And $S_1^2/2g$ represents the foot-pounds of energy carried over from the preceding row of blades.

Friction effects may be allowed for by introducing experimentally determined coefficients in the right-hand member of (490) as follows:

$$(491) \quad \frac{R_1^2}{2g} = \frac{S^2}{2g} = \frac{c_1 S_1^2}{2g} + c_2 JE.$$

¹ See D. A. Low, *Heat Engines*, p. 434.

The value of c_1 has been found equal to about 0.5, and c_2 equals 0.9 approximately.¹

Experience indicates also that the carry-over energy from one row of blades to the next is just about equal to the friction and leakage losses. Hence under actual conditions for a single row of blades

$$(492) \quad \frac{S^2}{2g} = \frac{R_1^2}{2g} = JE,$$

and

$$(493) \quad S = R_1 = 223.8 \sqrt{E}.$$

It will be recalled that this is the expression used to compute the spouting velocity of steam issuing from the nozzle of an impulse turbine.

By Newton's second law, the force equals mass multiplied by acceleration. Acceleration is the rate of change of velocity. Hence when steam starting from rest acquires a velocity of S feet in one second, S is also the acceleration. The tangential component of the force set up by steam impinging on the moving blades is $S \cos \alpha$. The tangential component for the steam leaving the blade is $S_1 \cos \gamma$. Then from the velocity diagram in Fig. 186,

$$(494) \quad S_1 \cos \gamma = R_1 \cos \alpha - S_b = S \cos \alpha - S_b.$$

The total force causing rotation of the moving blades is the sum of the forces set up by the entering and leaving steam. It equals

$$(495) \quad \frac{M}{g} (S \cos \alpha + S \cos \alpha - S_b).$$

The distance traveled by the force in one second is S_b , hence the work done equals

$$(496) \quad \frac{MS_b}{g} (2 S \cos \alpha - S_b).$$

For a flow of one pound of steam per second the total work per stage equals

$$(497) \quad \frac{2 S_b S \cos \alpha - S_b^2}{g}.$$

¹ A. Stodola, *Steam and Gas Turbines*, translated by L. C. Loewenstein.

The energy supplied by one pound of steam per second per stage by (492) is

$$(498) \quad \frac{2 S^2}{2g} = 2 J E.$$

The efficiency e for one stage then equals (497) $\div$ (498); that is,

$$(499) \quad e = \frac{2 S_b S \cos \alpha - S_b^2}{S^2}.$$

The value of e becomes maximum when

$$(500) \quad S_b = S \cos \alpha,$$

and the maximum value of e is

$$(501) \quad e_m = \cos^2 \alpha.$$

The angle α is usually about 20 degrees, hence at the point of maximum efficiency

$$(502) \quad S_b = 0.94 S,$$

and

$$(503) \quad e_m = 0.94^2 = 0.88$$

The number of stages theoretically desirable in a reaction turbine may be computed from (497).¹

$$(504) \quad Z \frac{2 S_b S \cos \alpha - S_b^2}{g} = 778 e E,$$

and

$$(505) \quad Z = \frac{778 e E g}{S_b^2 \left(2 \frac{S}{S_b} \cos \alpha - 1 \right)}.$$

For a given turbine the ratio S/S_b is about constant in all stages and the angle α is the same for all rows of blades. Hence (505) may be written

$$(506) \quad Z S_b^2 = B.$$

¹ See J. A. Moyer, *Steam Turbines*, and A. Stodola, *Steam and Gas Turbines* (translated by L. C. Loewenstein).

In the above equations

$$B = \text{a constant} = \frac{778 e E g}{2 \frac{S}{S_b} \cos \alpha - 1},$$

Z = the number of stages desirable.

For marine turbines the value of B in (506) varies from about 1.5×10^6 to about 1.6×10^6 . For large turbines the value of B varies from about 2.2×10^6 to 2.6×10^6 . The smaller values of B in each case indicates more refined design, more accurate machine work in the turbine construction, and greater care to reduce unequal expansion between the rotor and the shell to a minimum.

Reaction turbines consist of many rows of blades. The complete analytical design cannot be carried out correctly by considering each stage as detached from all the others as was done in the above equations. There is always a carry-in of energy from the preceeding stage which has to be considered. The flow through each stage is therefore not due solely to the pressure drop therein. Rational analysis requires that the blading be subdivided into groups consisting of rows of blades having approximately the same length. The energy equations for the flow of steam through a group of blades are not simple, and the method of computation is too long to be given a place in a general text on heat engines.¹

202. An Approximate Method of Computing Turbine Heat Rates. It is often advantageous to have available a quick, approximate method for determining turbine heat rates even though a complete analysis is to be made. Such a method was devised by Sir Charles A. Parsons for his reaction turbine; but since that time, the method has been found to apply about equally well to all types of turbines. The Parsons' method gives results which are sufficiently accurate for most purposes of preliminary estimation and may be used as an approximate check upon the

¹ See H. M. Martin, *Design and Construction of Steam Turbines*; John Morrow, *Steam Turbine Design*; A. Stodola, *Steam and Gas Turbines* (translated by L. C. Loewenstein).

results obtained from more exact but longer methods. This method is largely empirical. Parsons found that for all conventionally designed and built machines there is a definite relationship between the efficiency ratio, based on the Rankine cycle, and certain dimensions of the machine.

Let

A = a coefficient,

d = pitch diameter of bucket wheels or drums, in inches,

e_r = efficiency ratio at full load based upon the Rankine cycle,

N = revolutions per minute,

r = number of rows of moving buckets, or blades.

Then Parsons found that when

$$(507) \quad A = r \left(\frac{d}{10} \right)^2 \left(\frac{N}{100} \right)^2,$$

there is a surprisingly well-defined relation between the values of the coefficient A and the corresponding values of the efficiency ratio e_r . Points for a curve showing this relation are given in Table 60.

TABLE 60

e_r	A	e_r	A
.....		0.61	60 000
0.20	5 000	0.65	83 000
0.30	9 500	0.70	122 000
0.40	18 500	0.75	213 000
0.50	32 000	0.79	300 000

The values from this table require adjustment if the conditions under which the turbine operates or its proportions are abnormal. For turbines smaller than about 500 kw. rating, the efficiency ratio is less than indicated in the table due to the disproportionately large rotation and leakage losses.

The value of A for a turbine is determined largely by commercial factors. The cost of the turbine increases with the larger values of A . But for a considerable range in the higher

values of A the cost increases at a much lower rate than the efficiency. Hence over this range, a 20 per cent increase in the efficiency may be obtained for considerably less than a 20 per cent increase in the cost of the turbine.

For given operating conditions of steam pressure, superheat, vacuum, etc. the selection of a value of A is analogous to selecting the ratio of bucket velocity to steam velocity S_b/S . For impulse turbines the best theoretical value of the ratio was found to be about 0.47. This corresponds to a value of A of about 190,000. For reaction turbines the best velocity ratio is about 0.90 which corresponds to a value of A equal to about 350,000 for modern central station conditions of operation. Such high values of A are not found in practice however. The maximum value obtains for the Ljungstrom turbine,¹ for which the value of A is about 220 000. For other short-cut methods of computing turbine heat rates see the references below.²

203. Regenerative Cycle. For a given set of conditions no engine can be more efficient than the Carnot. It will be recalled

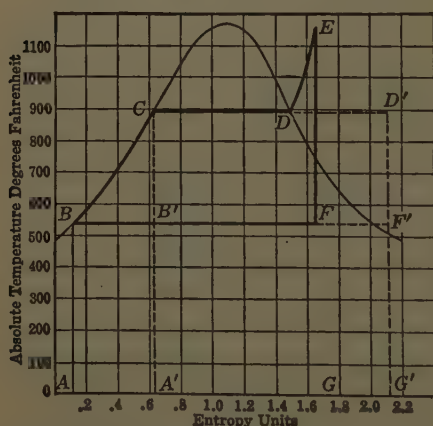


FIG. 187.—CARNOT AND RANKINE CYCLES WITH SUPERHEAT

that the Carnot cycle is represented on the TN -plane by a rectangular area as $B'CD'F'$, Fig. 187, for superheated steam in this instance. But the boundary lines of the rectangular area correctly represents the relations between temperature and entropy during the various steps of the cycle, and the area inclosed, correctly represents the quantity of heat transformed

¹ For description and data for the Ljungstrom turbine see *Ingenieur*, July, 1915; *Engineering* (London), Apr. 12, 1912, p. 482; May 4 and 11, 1923, pp. 542 and 577; Sept. 30, 1927, p. 411; and Oct. 21, 1927, p. 507. Also *Power*, Feb. 6, 1923, p. 229, A. Stodola, *Steam and Gas Turbines*.

² *Power*, Dec. 6, 1921; p. 895; and *Zeitschrift des Vereines deutscher Ingenieure*, Oct. 7 and 28, 1922, pp. 955 and 1025.

into useful work whether wet, dry and saturated, or superheated steam be supplied the engine or turbine. In this same figure, the area $BCDEF$ represents the quantity of heat transformed into work in an ideal Rankine engine when supplied with superheated steam. In this particular instance these two areas represent the quantities of heat transformed into work per pound of superheated steam supplied at an initial pressure of 375 pounds per square inch absolute and, carrying the same quantity of total heat in each case. In the case of the Rankine engine, however, the superheat is added at constant pressure and for the Carnot engine it is added at constant temperature. The efficiency of the Carnot cycle for these conditions is 40.3 per cent, while that of the Rankine is 33.9 per cent.

Professor Cotterill was one of the first to suggest that there are conceivable methods of bringing actual engine cycles to a closer approximation of the Carnot cycle than even the Rankine cycle.¹ One of these methods is the employment of what has come to be known as the regenerative cycle, which is illustrated in Figs. 188 and 189. It consists in heating the boiler feed water progressively in a series of heaters up to or near the boiling point at boiler pressure. Small quantities of steam for heating the feed water are extracted or bled from the engine or turbine at suitably different pressures during its expansion. The total steam extracted for regenerative heating in a series of from two to five stages varies from 10 to 20 per cent of the total steam flow at the throttle.² By this expedient high-grade heat energy is prevented, to a large extent, from becoming low-grade energy. This is the basic reason why the regenerative cycle is more efficient.

In 1898, the Nordberg Manufacturing Company built the first engine embodying the regenerative cycle. It was a quadruple expansion pumping engine, and was installed in the Wildwood Station, Pittsburgh, Pa.³ The regenerative cycle did not come into general use with steam engines, because with the small units, operating at relatively low steam pressures and low superheats,

¹ See Cotterill, *The Steam Engine*, second edition, 1890, pp. 235, 362, and 420.

² For methods of computing heat balances see Transactions, A.S.M.E., 1923, p. 713; and *Power Plant Engineering*, July 1, 1929, p. 739, and Aug. 1, 1929, p. 856.

³ See R. H. Thurston, *The Steam Engine at the End of the Nineteenth Century*, Transactions A.S.M.E., vol. 21 (1900), p. 181.

the small savings did not overbalance the additional first cost and the greater operating expense due to the added complication. It is only since about 1920 with the arrival of high initial pressures and temperatures that regeneration is being generally employed in modern central stations in which turbo-generator units with ratings of from 30 000 kw. to 200 000 kw. are being installed,

The temperature-entropy diagram for the 80 000 kw. regenerative unit No. 4 recently installed in the Hudson Avenue Station of the Brooklyn Edison Company is shown in Figs. 188a and 189.¹ Figure 189 is based upon the assumptions that steam

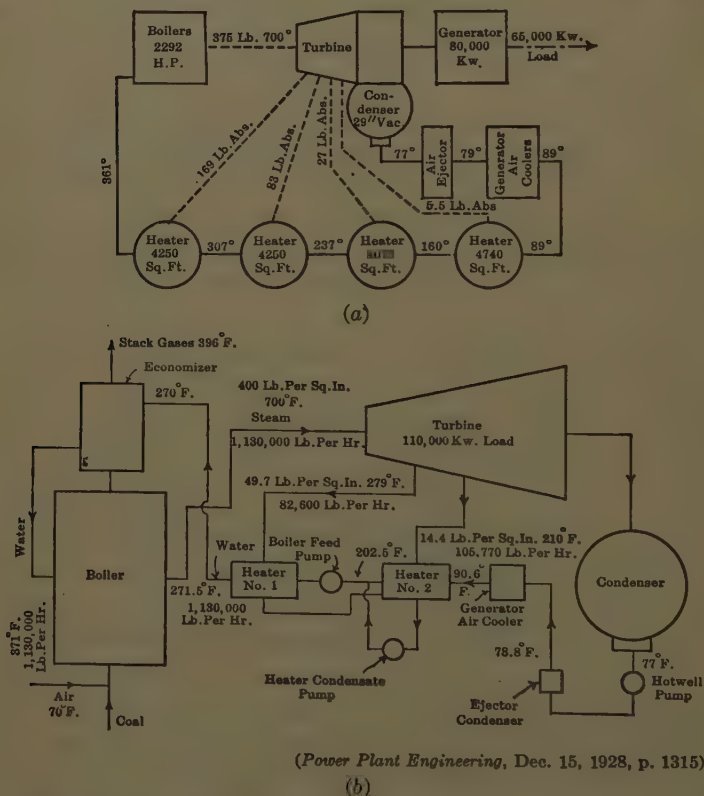


FIG. 188.—HEAT-CYCLE DIAGRAMS. (a), UNIT No. 4 HUDSON AVENUE STATION; (b), UNIT No. 5 HUDSON AVENUE STATION

expansion in the turbine between extraction points is strictly adiabatic, and that the heaters are ideal heat exchangers capable

¹ See Mechanical Engineering, April, 1927, p. 322.

of heating the feed water passing through them to the temperature of the extraction steam entering the respective heaters in a dry and saturated condition. Actually this is impossible, since the heaters are of the so-called *closed* type in which the steam and feed water are separated by metal walls similar to surface condensers, and some temperature head (5 to 10 degrees Fahrenheit), as may be determined from the pressures and temperatures shown in Fig. 188a, is necessary. In this particular installation there are a series of four heaters. Steam is extracted for the heaters at the four pressures indicated in the figure. Based upon the two assumptions stated above, the expansion of the steam through the turbine follows the zigzag line *Eyxwvutsrq* on the temperature-entropy diagram of Fig. 189. If there were an infinite number of heaters the expansion line would follow the broken-line curve *JK* which is parallel to the liquid line *BC*. In that case the useful work and the ideal efficiency would equal the values for the corresponding Carnot cycle.

For the conditions of pressures and temperatures shown in Fig. 188a, the efficiencies of the Carnot, Rankine, and regenerative cycles are found to be as follows.

Initial pressure $p_1 = 375$ lbs. per sq. in. abs.

Initial temperature of superheated steam $t_s = 700^\circ \text{ F.} = 1160^\circ \text{ F. abs.}$

Temperature of saturated steam t_1 at 375 lbs. = $438^\circ \text{ F.} = 898^\circ \text{ F. abs.}$

Temperature of exhaust steam $t_2 = 77^\circ \text{ F.} = 537^\circ \text{ F. abs.}$

Corresponding exhaust pressure $p_2 = 0.459$ lbs. per sq. in. abs.

For steam superheated at constant temperature, the Carnot efficiency is

$$e_c = \frac{T_1 - T_2}{T_1} = \frac{t_1 - t_2}{T_1} = \frac{438 - 77}{898} = 0.402$$

From the rectangular area *B'CD'F'* of Fig. 189 which represents the same number of B.t.u. as was supplied per pound of steam to the actual turbine, the efficiency of the Carnot cycle equals

$$e_c = \frac{\text{area } B'CD'F'}{\text{area } A'CD'G'} = 0.402$$

For the Rankine cycle with steam superheated at constant pressure, the value of the efficiency is

204. Reheat Cycle. Due to the fact that rotation losses in steam turbines increase rapidly as the percentage of moisture increases in the expanding steam (see Fig. 181), it has been found profitable to return the entire quantity of steam flowing through a turbine to a reheat boiler after its partial expansion through the first few stages of a turbine, or after expansion is completed in the first unit of a compound turbine. The reheat cycle is shown on the TN -plane in Fig. 190. The area $KFGJ$ represents the heat added in raising the superheat to the initial value before completing the expansion. For usual conditions, it is economically profitable to employ the reheat cycle only when the initial pressure is 600 pounds per square inch or higher. Fig. 190 shows the various heat areas for a combined reheat and regenerative cycle. The ideal efficiency of this cycle is

$$e = \frac{\text{area } BCDEFGHL}{\text{area } ABCDEFGHJ'}$$

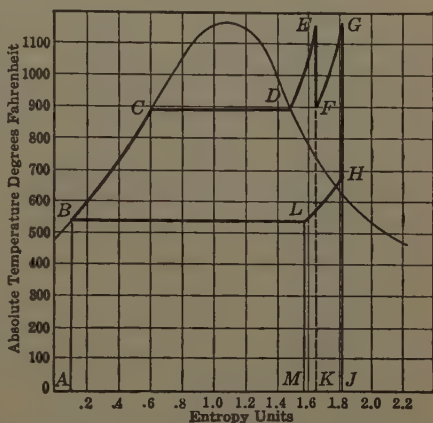


FIG. 190.—COMBINED REHEAT AND REGENERATIVE CYCLES

There are some practical objections to reheating as follows: (1), increased cost and complication of the plant; (2), storage of energy in the reheating system which usually necessitates a special valve on the low-pressure section of the turbine to prevent overspeeding. The most recent tendency is toward using live-steam reheaters placed beside the turbine. This would restrict the reheat-temperature rise to a relative small range but would greatly simplify the plant.¹

¹ Several heat-cycle diagrams for large modern installations are shown in *Power Plant Engineering*, Dec. 15, 1929, p. 1314; Nov. 1, 1929, p. 1174; Nov. 15, 1929, p. 1208; and Dec. 15, 1929, p. 1351.

205. Binary-Vapor Cycles. Carnot showed that the thermal efficiency of any heat engine depends only upon the temperature range through which the substance used as a carrier of heat passes as the engine cycle is traversed. As has already been shown, there is no perfect gas, no 100 per cent perfect engine, no perfect anything in nature, seemingly, so there is no perfect medium which may be used as a carrier of energy in any heat-engine cycle. For practical reasons some substances are better than others. Hence from time to time, the use of substances other than H_2O vapor have been proposed, among them being alcohol, ammonia, ether, sulphur dioxide, sulphur, benzophenone, and diphenyl oxide.¹

The chief disadvantages of water vapor are that the pressures accompanying saturated temperatures above $500^\circ F.$ become inconveniently high, and at low temperatures such as those available for condensing water (say $40^\circ F.$ to $100^\circ F.$) the corresponding pressures are much below atmospheric. At low pressures accompanied by correspondingly low temperatures, the volumes of water vapor are very large. These facts of nature necessitate the employment of large, heavy, and expensive equipment in order to realize actual efficiencies approaching reasonably close to those of the ideal Carnot for corresponding temperature limits. To meet these objectionable features and at the same time extend the available temperature range Professor E. Josse of the Royal Technical High School of Charlottenberg proposed, in 1899, the use of the binary-vapor cycle.² In this combination high-pressure steam was supplied to the first cylinder of a compound unit. The exhaust steam at a vacuum of about 24 inches of mercury was utilized to evaporate sulphur dioxide at a temperature of about $140^\circ F.$ and a pressure of 156 pounds per square inch. The SO_2 vapor was utilized in the second cylinder of the compound engine to develop more power after which it was exhausted to a

¹ See R. H. Thurston, *The Series-Vapor and Heat-Waste Engines*, Journal of the Franklin Institute, Oct.-Dec., 1902, pp. 291, 367, 447.

H. H. Dow, *Diphenyl Oxide Bi-fluid Power Plants*, Mechanical Engineering, Aug., 1926, p. 815.

² See R. H. Thurston, *Sulphur-Dioxide and the Binary Engine*, Journal of the Franklin Institute, June, 1903, p. 429.

R. H. Thurston, *Manual of the Steam Engine*, Part I, p. 704, 1907. Electrical World and Engineer, vol. 88, 1901, p. 219.

second condenser at a temperature of 60° F. and a pressure of 40 pounds per square inch. Higher steam pressures and the development of the steam turbine, in which very low exhaust pressures and the correspondingly large steam volumes could be utilized advantageously, prevented the successful economic development of the binary cycle proposed by Josse.

In 1913, W. L. R. Emmet¹ began an investigation which led to the development of a binary system in which mercury vapor is employed in a turbine similar to steam turbines to utilize temperatures higher than is considered possible with steam (H_2O vapor). The mercury condenser serves also as a boiler in which steam is generated, and the ordinary steam turbine is the second prime mover of this binary system. Thus a larger temperature range than is usually available with steam alone is realized without developing excessive pressures. For instance, the boiling point of mercury at a pressure of 100 pounds per square inch absolute is about 907° F.² At an absolute pressure of 0.5 pounds the boiling point is 415° F. Allowing no temperature drop in a theoretically perfect mercury-condenser-steam-boiler unit, saturated steam at a pressure of 292 pounds per square inch absolute may be generated. The steam condenser may be readily operated at an absolute pressure of 1.0 inch of mercury which corresponds

to a temperature of 79° F. The heat quantities involved in such an ideal binary system may be most readily pictured on the TN -plane of Fig. 191. The various efficiencies realizable from this

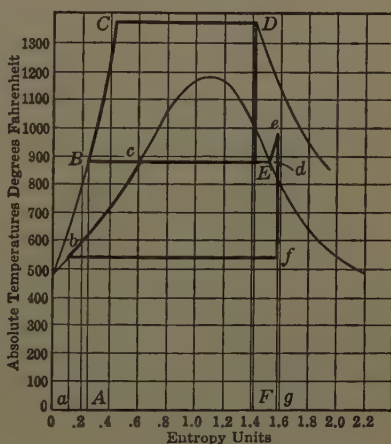


FIG. 191.—BINARY CYCLE FOR MERCURY AND WATER VAPORS

¹ See Emmet, *Mercury-Vapor Process*, Transactions A.S.M.E., 1924, pp. 253–285.

² L. A. Sheldon, *Properties of Mercury Vapor*, Transactions A.S.M.E., 1924, p. 276.

system may also be readily computed by reference to the above-mentioned figure and vapor tables for mercury and steam as carried out below for the conditions stated above.

The Carnot efficiency for the system when operating between the extreme temperature limits cited is

$$e_c = \frac{T_1 - T_2}{T_1} = \frac{1366.5 - 538.7}{1366.5} = \frac{827.8}{1366.5} = 60.6 \text{ per cent.}$$

The Rankine efficiency for the mercury plant alone is

$$\begin{aligned} e_m &= \frac{H_1 - h_2 - T_2(n_2 - n_1)}{H_1 - h_2} \\ &= \frac{149.7 - 14.3 - 875(0.1238 - 0.0215)^1}{149.7 - 14.3} = \frac{149.7 - 103.9}{135.4} \\ &= \frac{45.8}{135.4} = 33.8 \text{ per cent.} \end{aligned}$$

The Rankine efficiency for the steam plant alone is

$$e_s = \frac{H_1 - H_2}{H_1 - h_2}.$$

Taking the appropriate values from the Mollier chart, and vapor tables for steam superheated 100 degrees, one finds

$$e_s = \frac{1266 - 850}{1266 - 47} = \frac{416}{1219} = 34.2 \text{ per cent.}$$

In the mercury-water binary system, it is assumed that all the heat generated directly from fuel is used under the mercury boiler and that saturated steam is generated only by the heat rejected from the mercury turbine. The surplus heat in the mercury-boiler stack gases is used to superheat the steam 100° F., and to preheat the air for combustion in the mercury-boiler furnace. The feed water entering the steam boiler is assumed to be at the temperature in the steam condenser.

The heat rejected in each pound of mercury leaving the mercury turbine as computed in evaluating e_m above equals

$$T_2(n_2 - n_1) + h_2 = 103.9 \text{ B.t.u.}$$

Under these assumed conditions the generation of each pound of

¹ These values are taken from mercury vapor tables in Transactions A.S.M.E., 1924, p. 276.

saturated steam requires 1203 B.t.u. Then for each pound of steam generated

$$\frac{1203}{103.9} = 11.6$$

pounds of mercury must be circulated. Hence the *TN*-diagram for mercury and water of Fig. 191 is constructed for 11.6 pounds of mercury and for one pound of water.

The ideal efficiency of the combined binary cycle outlined above and for the conditions stated equals

$$\begin{aligned} e_b &= \frac{\text{B.t.u. converted into useful work}}{\text{B.t.u. supplied}} \\ &= \frac{45.8 \times 11.6 + 416}{135.4 \times 11.6} = 60.0 \text{ per cent.} \end{aligned}$$

From the corresponding areas in Fig. 207

$$e_b = \frac{\text{area } BCDE + \text{area } bcdef}{\text{area } ABCDF}.$$

If the efficiency ratio of the mercury and steam turbo-generators is assumed to be 80 per cent, and if the efficiencies of the mercury boiler is 85 per cent and the mercury-condenser-steam-boiler is assumed to be 97 per cent, the plant thermal efficiency would then be

$$0.80 \times 0.85 \times 0.97 \times 0.600 = 0.396 = 39.6 \text{ per cent.}$$

A kilowatt-hour on that basis could be delivered to the switchboard for

$$\frac{3412}{0.396} = 8615 \text{ B.t.u. per hr. supplied in the fuel.}$$

The efficiencies stated above are about as high as can be realized conveniently with modern equipment operating on the mercury-water cycle. The best modern high-pressure steam plants operating on combined reheat and regenerative cycles are capable of delivering a kilowatt-hour at the switchboard for an expenditure of about 11 000 B.t.u. in the fuel.

In view of the fact that any binary system involves additional fixed charges and operating costs for the two practically separate plants; that it necessitates the use of an expensive substance (compared to water); that the proper coördination of the operation of the two plants involves complication; that the depreciation and maintenance charges must be higher; and that the possible im-

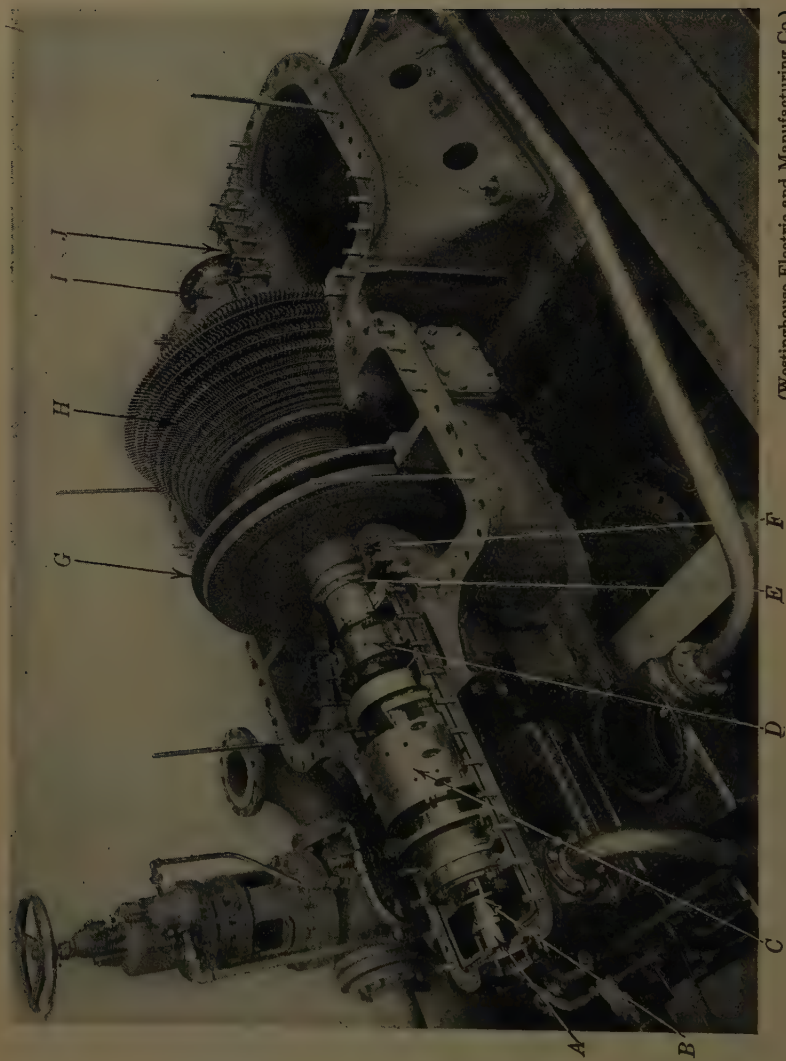


FIG. 192.—WESTINGHOUSE IMPULSE-REACTION TURBINE 500 TO 8 000 KW.
(Westinghouse Electric and Manufacturing Co.)

provement in thermal efficiency over that for steam alone is no greater than indicated above, it would seem that it remains to be demonstrated that any binary system can become an economic success.

206. Modern Reaction Turbines. Figure 192 is an interior view which is typical of Westinghouse impulse-reaction turbines built in sizes ranging from 500 kw. to 8000 kw., for initial steam pressures varying from 100 to 300 lbs. per square inch and for rotative speeds from about 1200 to 1800 r.p.m.

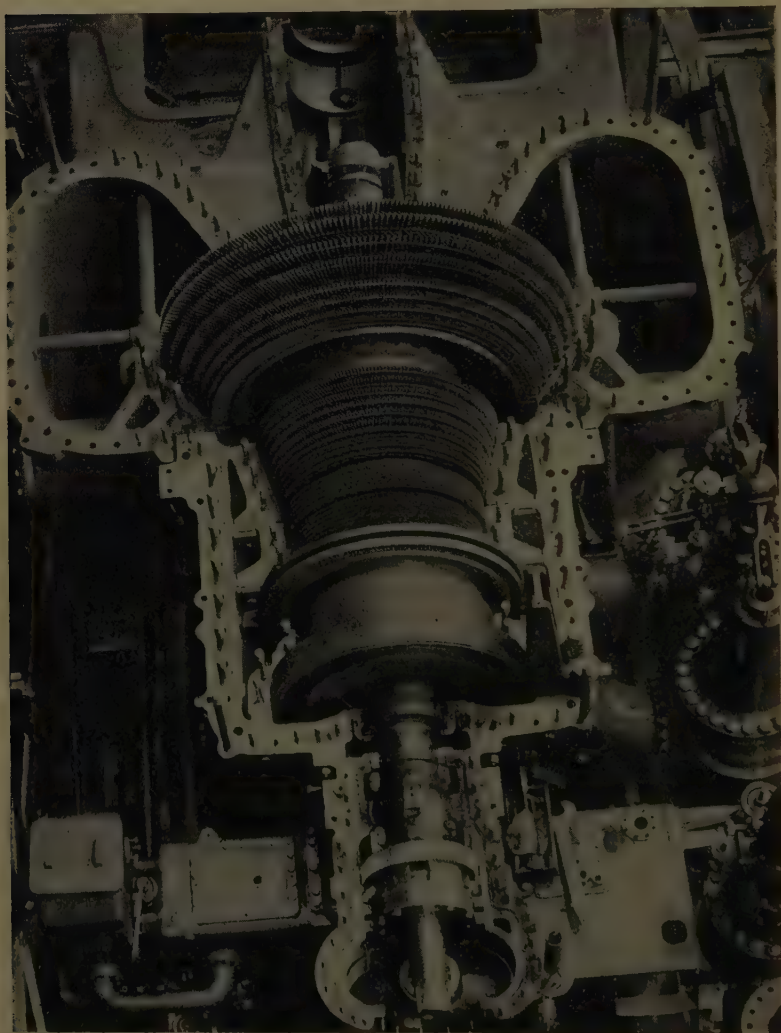
This arrangement of turbine elements, consisting of double-bucket-row impulse wheels (usually one wheel comprising one impulse stage) at the high-pressure end followed by Parsons reaction elements was developed by George Westinghouse. Patents on this combination were first granted him in 1903. This combination permits a more stable and cheaper construction than is possible in the reaction machine, due to its being much shorter. This style of turbine is especially advantageous for the smaller sizes.¹

A spring-controlled weight at *A* on the main shaft is automatically thrown out by centrifugal force releasing a trigger which in turn releases a spring-loaded stop-valve (not shown). The stop-valve shuts off the steam supply to the turbine in case its speed accidentally rises more than 10 per cent above normal. A centrifugal-pump runner is located at *B* which circulates the lubricating oil at a pressure of from 10 to 40 pounds per sq. inch through all the turbine bearings and back to a filter and storage tank. The Kingsbury thrust-bearing is indicated by *C*, the outboard turbine-bearing is at *D*. At *E* is another centrifugal-pump runner which serves as a water seal to prevent leakage of steam from the casing around the shaft; in addition there is a labyrinth steam-seal packing at *F*. The water seal and labyrinth packing are both necessary at the high-pressure end in order to completely eliminate steam leakage. A single pressure stage of impulse blading, consisting of two velocity stages (double-bucket row) at *G*, receives the high-pressure steam as it enters the turbine. Then the steam passes into the reaction blading indi-

¹ See H. T. Herr, *Recent Developments in Steam Turbines*, Journal of the Franklin Institute, Feb. and Mar., 1913, pp. 91 and 273.

cated by *H*. There are ten stages of reaction blades enclosed in a conically bored housing. The inboard bearing is indicated by *I* and the turbine end of the shaft coupling is shown at *J*.

Fig. 193 is an interior view of the larger sizes of Westinghouse



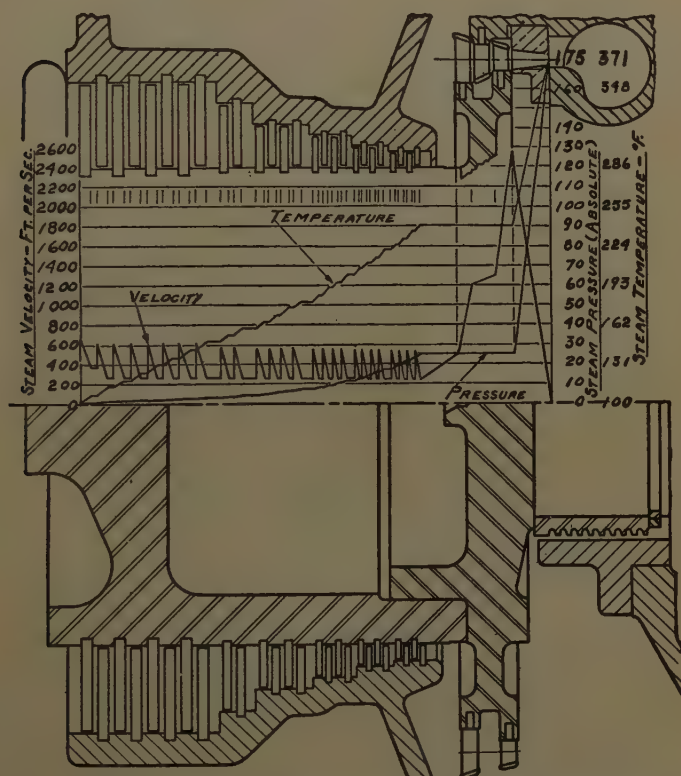
(Westinghouse Electric and Manufacturing Co.)

FIG. 193.—IMPULSE-REACTION TURBINE FOR 10 000 KW. AND UP

impulse-reaction turbines (10 000 to 65 000 kw.)¹ It embodies

¹ See *Electric Journal*, Nov. 1928, pp. 534-537.

the same general features as shown in the smaller turbines typified by Fig. 192. The chief differences being in the larger number of reaction stages (30 in this particular machine), and the moving blades are attached to a conical drum having two different diameters.



(Westinghouse Electric and Manufacturing Co.)

FIG. 194.—TEMPERATURE PRESSURE AND VELOCITY CURVES FOR IMPULSE-REACTION TURBINE

Temperature, velocity, and pressure curves for the steam as it progresses through the stages of a 500-kw. impulse-reaction turbine, are shown in Fig. 194. The abscissas for these curves are the successive turbine stages shown in conventionalized outline.

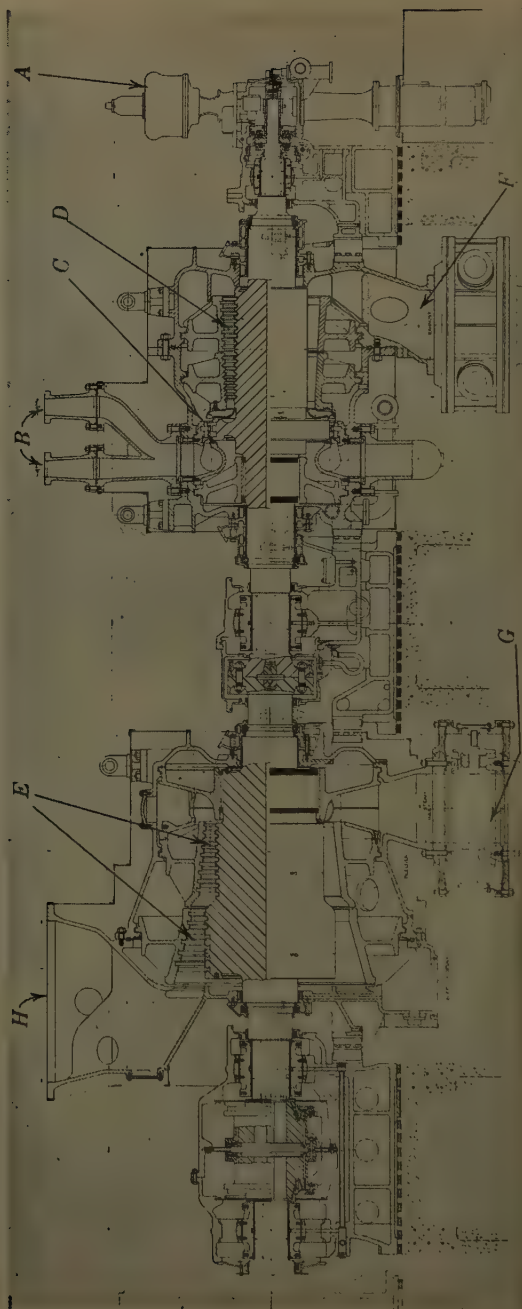
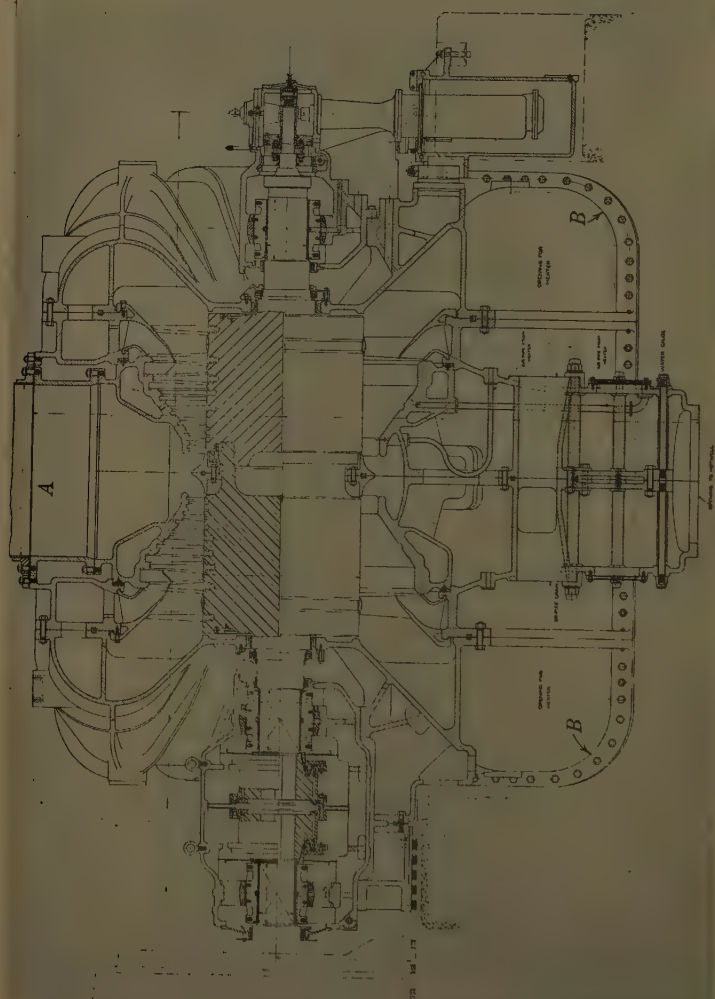


Fig. 195.—HIGH-PRESSURE END OF TANDEM CROSS-COMPOUND 50 000 KW. TURBINE
(Westinghouse Electric and Manufacturing Co.)

Longitudinal sections of a typical design of Westinghouse tandem, cross-compound turbines are shown in Figs. 195 and 196. This general style of turbine is built in sizes varying from 40 000 kw. to 200 000 kw. The high and intermediate sections, Fig. 195, are coupled tandem and enclosed in separate cast-steel cylinders. The low-pressure section, Fig. 196, compounded with them, is mounted on a separated shaft, and housed in an entirely separate cast-iron cylinder. The low-pressure section, with its generator direct-connected, may be operated at the same speed or at a different speed from the tandem section. The high and intermediate sections have solid, forged-steel spindles or drums mounted in three main bearings, and they are coupled rigidly together. The low-pressure rotor is of the semisolid type of construction mounted on two main bearings.

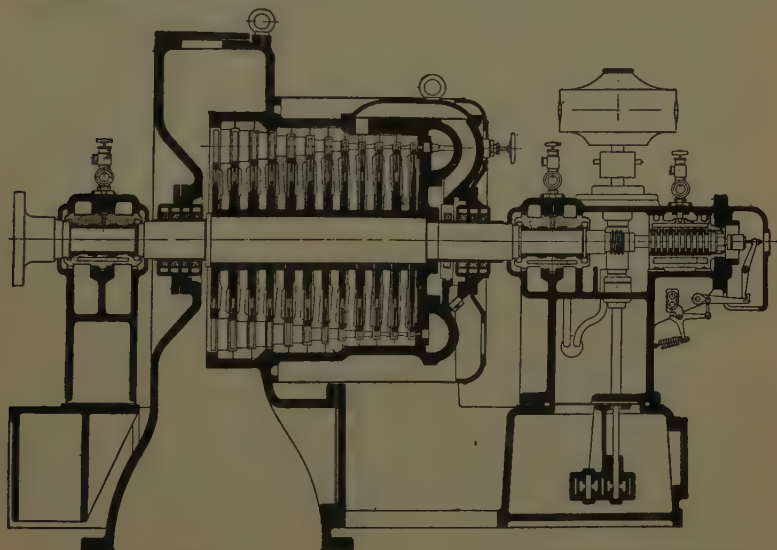
The multicylinder type of construction originated in the British experimental yacht *Turbinia* in 1896. This type of turbine has continued in use for marine propulsion continuously since that date. It was employed to a limited extent for land service in the early 1900's and was revived about 1914. It is employed alike for impulse turbines and reaction turbines in all sizes larger than about 50 000 kw. The multicylinder construction is very convenient, and it is the most logical one for all large turbines operating on the reheat cycle.

In Fig. 195, *A* is the speed governor which controls steam admission to the high-pressure end only. *B* indicates the high-pressure steam inlets, divided into two branches in order to give greater flexibility to the piping, and thus reduce mechanical strain on the turbine cylinder due to high temperatures. A double-bucket row of impulse blading is indicated at *C* which is followed by 15 stages of reaction blading *D* (in this particular instance). The steam leaves the high-pressure section at *F* on its way to a reheat boiler. It enters the intermediate section at *G* and passes through the 15 stages of reaction blades *E* mounted in the conically-bored cylinders. The steam then passes from *H* directly over to the inlet of the low-pressure section at *A* (Fig. 196). In this section the steam is divided and flows through the blading in both directions along the shaft. This type of con-



struction accomplishes two essential results: the end thrusts on the rotor are balanced or nearly so; and the required areas, which permit a reasonably small velocity of flow of the extraordinarily large volumes of steam in the low-pressure stages are provided. One of the difficult problems in designing large turbines is the proper proportioning of the low-pressure end for efficient energy extraction without exceeding safe limits of structural design. One feature of this problem is to properly brace the very large low-pressure cylinder to withstand the external pressure of the atmosphere since the internal pressure is frequently less than 0.5 pound per sq. in. abs. The exhaust flanges of the turbine cylinder are connected directly to the condenser placed just below the low-pressure cylinder.¹

207. Modern Impulse Turbines. A longitudinal, vertical section of a De Laval multistage turbine is shown in Fig. 197. This style of turbine is built in sizes ranging from 500 kw. to



(De Laval Steam Turbine Co.)

FIG. 197.—SECTION THROUGH DE LAVAL MULTISTAGE TURBINE

¹ For a more detailed description of large reaction turbines see *Power*, Nov. 30, 1926, p. 815, for 80 000 kw. Westinghouse unit; *Power Plant Engineering*, Sept. 1, 1927, p. 940, for 50 000 kw. Allis-Chalmers unit; *Power Plant Engineering*, Oct. 15, 1928, p. 1077; 104 000 kw. Westinghouse Turbine at Crawford Ave. Station, Chicago; *Power*, Feb. 5, 1929, p. 235, for 160 000 kw. Westinghouse unit at Hell Gate Station, New York.

5000 kw. It consists of a series of single-stage, single-bucket row wheels mounted compactly on a short rigid shaft. (The De Laval wheels of the older turbines were mounted on small flexible shafts which permitted the rotating element to revolve about its center of mass. It often happens that the center of mass and the geometric center do not coincide exactly.) Each wheel revolves in a separate chamber formed by diaphragms held transversely in the cylindrical casing. Steam enters the steam chest at the right-hand end of the casing, and flows successively through the nozzles and buckets of each stage, and finally into the exhaust at the left



(General Electric Co.)

FIG. 198.—3 000 HP. CURTIS HIGH-EFFICIENCY TURBINE

which is usually connected directly to the condenser inlet below (not shown). Steam leakage from stage to stage is reduced to a minimum by labyrinth packings in each diaphragm through which the shaft passes. The end thrust on the turbine shaft is small because of the symmetric form of the rotating buckets. But to maintain proper axial clearances an adjustable, marine-type, thrust bearing is provided. It is shown in Fig. 197 near the extreme right-hand end of the turbine.

Figure 198 is an exterior view of a 3000-hp. Curtis high-efficiency turbine driving two centrifugal pumps through a reduction gear. This style of turbine is built in sizes ranging from 1000 to 6000 kw. The general form of rotor used in this type of turbine is shown by Fig. 199. Sometimes, when the first cost must be limited, the number of stages employed is small and the

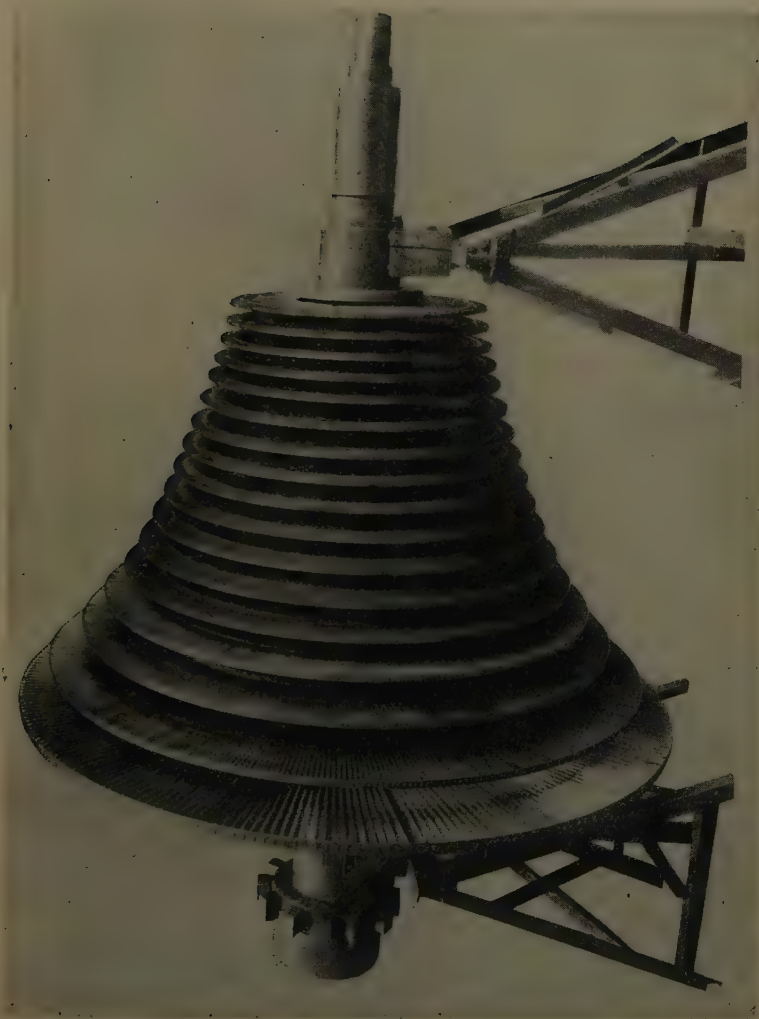
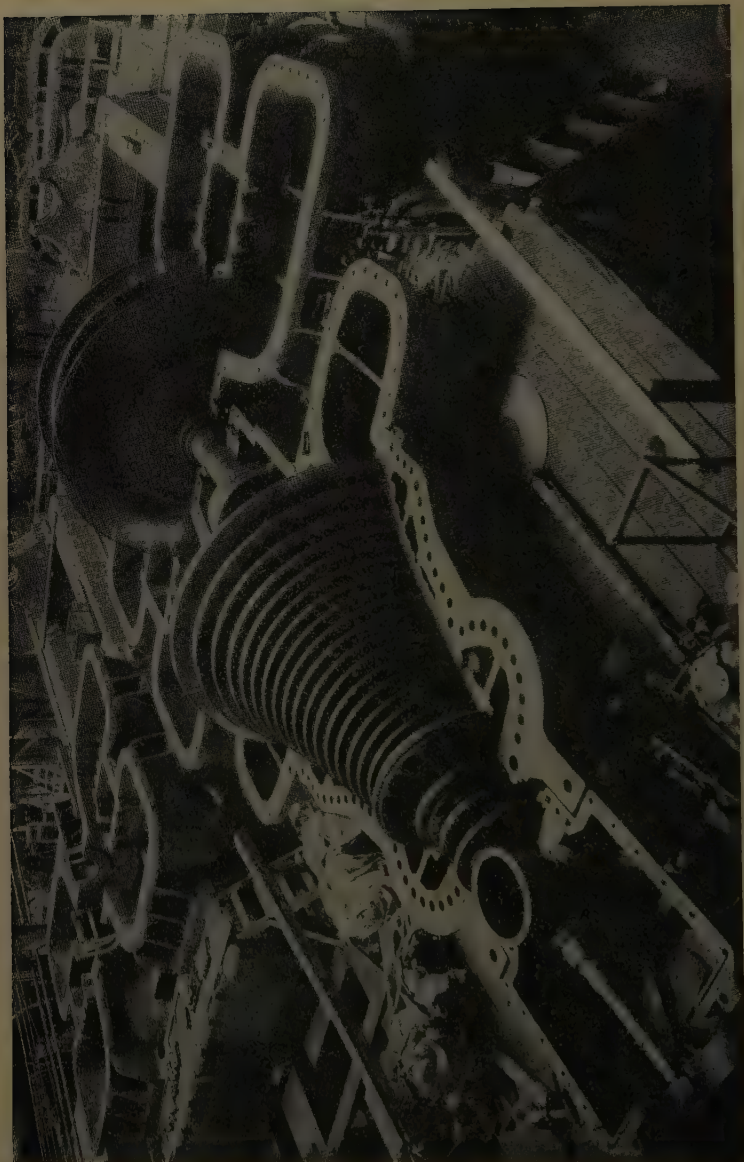


FIG. 199.—ROTOR FOR 17-STAGE CURTIS HIGH EFFICIENCY TURBINE
(General Electric Co.)



(General Electric Co.)

Fig. 200.—50 000 Kw. 20-Stage Tandem-Compound Curtis Turbine

first stage consists of a double-bucket-row wheel. Ordinarily the number of stages employed varies from 15 to 30. When an extremely low steam rate is most essential, turbines of this style containing up to a maximum of 40 stages have been found profitable.¹

Fig. 200 shows an interior view of a 50 000 kw. Curtis turbine installed in the Richmond Station of the Philadelphia Electric Company in 1925. It is tandem-compound with double flow (*i.e.* both ways axially) in the low-pressure cylinder in order to provide larger exit areas for the exhaust steam. This turbine was designed to operate with steam at an initial pressure of 375 lbs. per sq. in. gage, superheated 230° F., and with an exhaust pressure of one inch of mercury absolute.

Figure 201 is a sectional view of a 94 000 kw. 1500 r.p.m. tandem-compound Curtis turbine which was first put into service in Long Beach Plant No. 3 of the Southern California Edison Company in July, 1928.

The speed governor *G* is driven by a worm gear on the main shaft. *R* is a steam-cylinder relay controlled by the governor. The relay exerts the necessary force to control the valves *V*. The valves *V* in turn control the number of first-stage nozzles which are open. There are 18 high-pressure stages as indicated by *H*. The three double-flow, low-pressure stages are indicated by guide line *L*. The main generator is rated at 90 000 kw. at a power factor of 90 per cent. An auxiliary, house generator of 4000 kw. capacity is coupled to the shaft of the main generator. The total length of the complete unit is 103 feet. The shaft is 16 inches in diameter. The total weight of the turbo-generator unit exclusive of auxiliaries such as condensers, pumps, etc. is 1 650 000 lbs. or about 17.5 lbs. per kw.

The turbine operates on the regenerative cycle. Steam is supplied to the turbine at a pressure of 415 lbs. absolute and 750° F. Steam is extracted for feed-water heating from the 5th, 10th, 14th, and 18th stages. The corresponding full-load pressures are respectively 229, 100, 36, and 7.5 lbs. absolute. At full load 1 037 000 lbs. of steam per hour enter the turbine.

¹ See S. A. Moss, *High Efficiency in 1000 kw. to 6000 kw. Steam Turbines*, Power, May 19, 1925, p. 775.

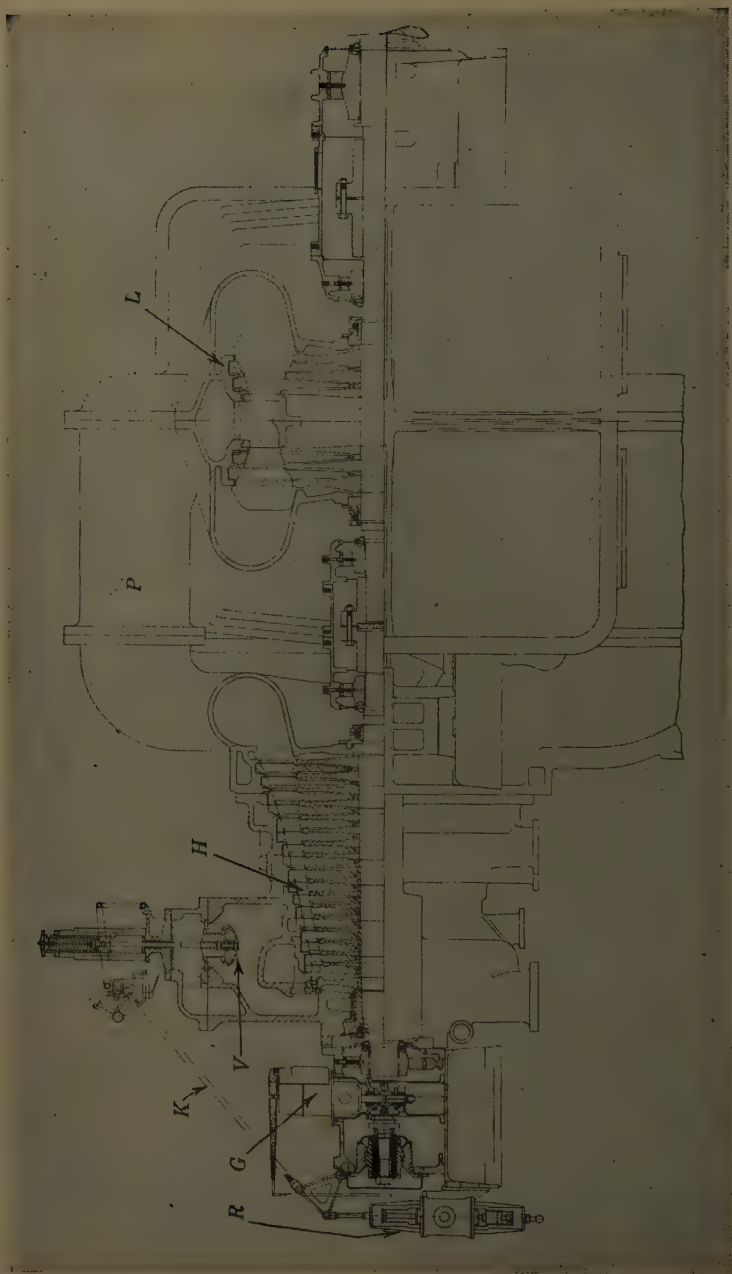


Fig. 201.—94 000 Kw. 21-Stage Tandem-Compound Curtis Turbine

(General Electric Co.)

FIG. 202.—
35 000 Kw.
20-STAGE
CURTIS
TURBINE-
GENERATOR



(General Electric Co.)

The maximum thermal efficiency based on station send-out occurring at 110 per cent rated load has been reported as about 27.5 per cent.¹ This is equivalent to 12 400 B.t.u. per kw.-hr. of send-out or to about 9800 B.t.u. per kw.-hr. for the turbine and generator alone.

Fig. 202 is a view of a 35,000-kw. 20-stage single-cylinder turbine-generator as installed in the Long Beach Plant No. 2 of the Southern California Edison Company. The turbine is designed for steam extraction from the 11th, 15th, and 18th stages for feed-water heating. The initial steam pressure at the throttle is 350 lbs. gage at a temperature of 700° F., and the exhaust pressure is 1.5 inches of mercury absolute. With extraction, the unit has produced, under test, a kw.-hr. at the switchboard for each 11 000 B.t.u. supplied at the throttle, or a thermal efficiency of about 31 per cent.

The main 35 000 kw.-11 000-volt generator, a 2800-kw. 2300-volt auxiliary generator, and a 45-kw., 250-volt exciter are all direct-coupled to the turbine shaft. This arrangement provides auxiliary power at a generating efficiency equal to that of the main unit without electrical connection with the main high-voltage switchboard. The extreme tip end of the exciter is just visible in the figure.²

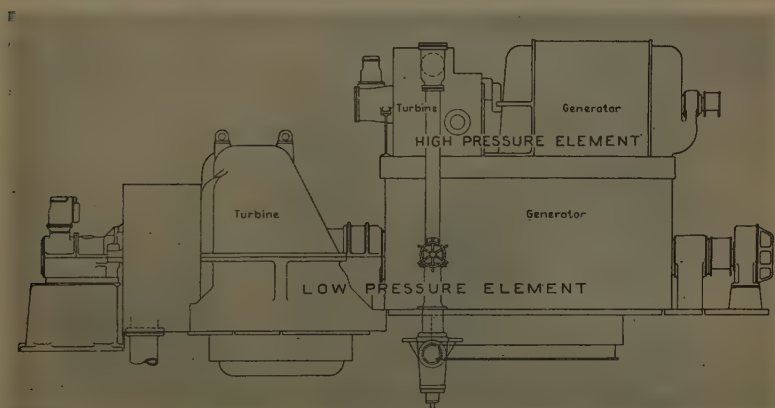
A high-pressure (1200 pounds per square inch) vertical-compound Curtis turbine-generator is shown in side elevation in Fig. 203. The vertical arrangement of the two elements of the compound unit saves floor space. Units of this general design are constructed to develop from 50 000 kw. to 125 000 kw.

A view of the lower half of the shell and rotor of a 14-stage high-pressure element is seen in Fig. 204. The various parts of the rotor are indicated by letters as follows: *A* is the worm gear for governor drive; *B*, end-thrust bearing; *C*, high-pressure main bearing; *D*, high-pressure labyrinth packing; *E*, first-stage, double-bucket-row wheel; *F*, second-stage, single-bucket-row wheel; *G*, low-pressure labyrinth packing; and *H* is the main bearing between the turbine and the generator.

¹ Power Plant Engineering, Apr. 1, 1929, p. 407.

² For complete description of this plant see *Power*, Feb. 17, 1925, p. 246.

The labyrinth packings consist of many rectangular grooves cut on the shaft (about 60 on the high-pressure end). Corresponding sharp-edged ridges are cut in rings, and held stationary in the frame. The annular clearance between the fixed and rotating parts is made as small as possible without bringing them into actual contact. The clearance distance is generally about 1/100 of an inch. The tortuous path thus provided for the flow of steam produces so much turbulence and resistance that only about one per cent of the total steam supplied escapes at the open end of the bearing. The leakage is not lost but is conducted to one of the succeeding stages of the turbine. The packings between the stages of the high-pressure element are also of the labyrinth style.



(General Electric Co.)

FIG. 203.—1200-LB. VERTICAL-COMPOUND TURBINE-GENERATOR

This 1200-pound turbine is operated on the regenerative and reheat cycles, there being from 3 to 5 stages of feed-water heating and one stage of reheating. In the 53 000 kw. Deepwater installation,¹ the steam enters the high-pressure element at 1200 lbs. gage and 750° F. It leaves at about 400 lbs. absolute and is taken to a reheat boiler where the temperature is raised to 750° F. again. The steam then enters the low-pressure element at a pressure of about 388 lbs. absolute.

¹ See N.E.L.A. Publication No. 289-73, June, 1929, for heat-cycle diagram.

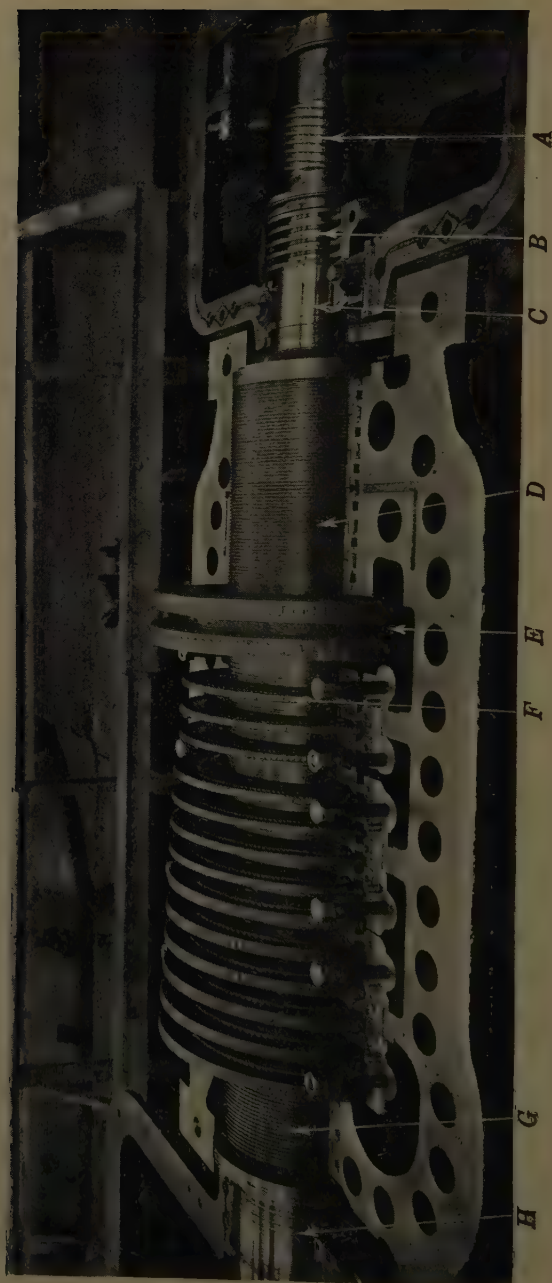


FIG. 204.—HIGH-PRESSURE ELEMENT OF A 1200-LB. TURBINE
(General Electric Co.)

Steam for feed-water heating is bled from four points as follows: exhaust of the high-pressure element, and at the 6th, 10th, and 15th stages of the low-pressure element. The respective absolute pressures and saturated temperatures at these points are:

Pressures 400, 138.5, 46.2, and 4.45 lbs.

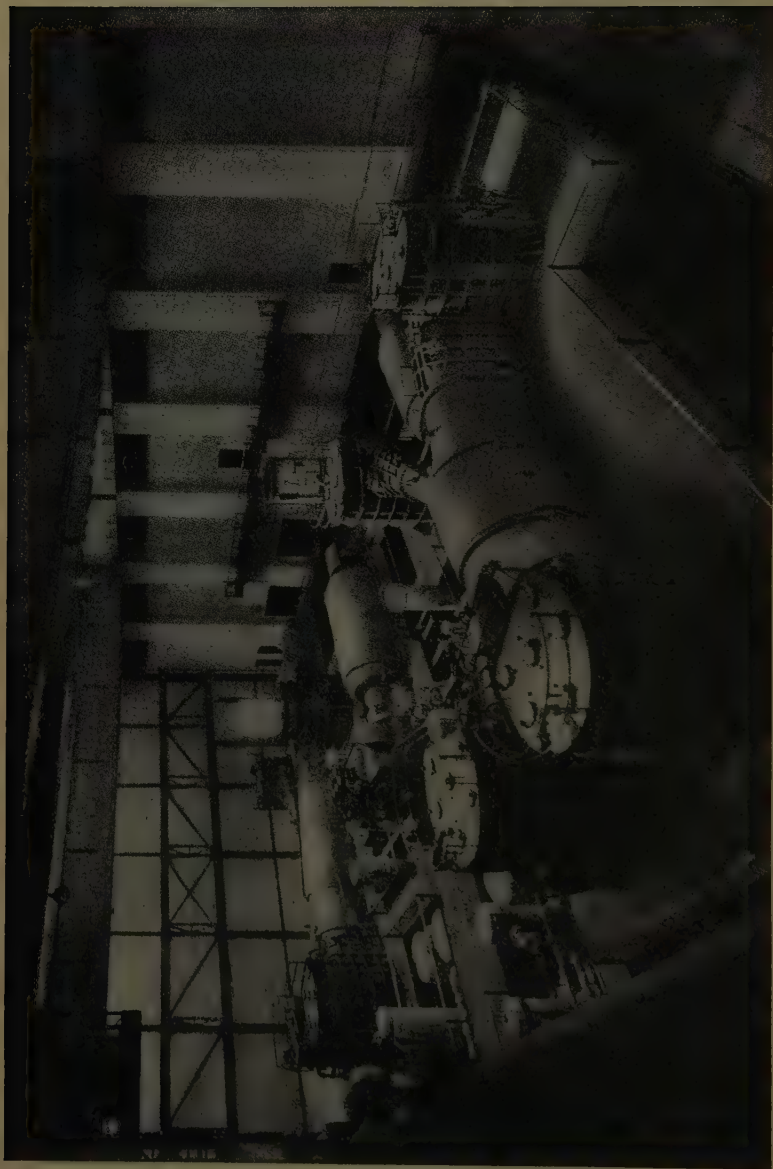
Temperatures 444.6, 352, 276, and 158° F.

The condensate is returned to the boiler at a temperature of 434° F. The best heat rate for this cycle is estimated to be **9235 B.t.u. per kw.-hr.** delivered by the generator at a load of 39 380 kw. This heat rate does not include the power to drive the auxiliaries. Allowing a boiler-room efficiency of 82 per cent and 6 per cent of the power generated to drive the auxiliaries, **a kw.-hr. may be delivered to the line for each 12 000 B.t.u. in the fuel.** This corresponds to a plant thermal efficiency of about 28.4 per cent.

A view of the three-element 208 000 kw. cross-compound unit recently placed in service in the State Line plant, located on Lake Michigan, is shown in Fig. 205. It is owned jointly by the Commonwealth Edison Co., the Public Service Co., the Northern Indiana Public Service Co., and the Interstate Public Service Co. It will be operated by the four companies on base load, furnishing power to each company proportional to its investment.

The three-element unit will be operated as a two-unit machine. It consists of a high-pressure 76 000 kw. generator seen in the center of Fig. 205, flanked by two low-pressure 62 000 kw. machines. The additional 8000 kw. is made up by a 4000 kw. house generator coupled to the shaft of each low-pressure generator. All three elements operate at 1800 r.p.m. Each low-pressure element is of the double-flow type and is equipped with four vertical condensers. The condensers are set vertically for the more convenient removal of tubes and less floor area is needed.

The steam is supplied by six boilers. It enters the high-pressure element at 650 lbs. gage and 750° F. It leaves at 72 lbs. absolute going to steam-coil reheaters. It enters the low-pressure elements at 71 lbs. absolute and 500° F. Steam for feed-water heating is extracted at five points at the following absolute pressures: 231, 140, 73, 30.3, and 5.9 lbs. Operating with a 29-



(Power Plant Engineering.)

FIG. 205.—208 000 KW. CROSS-COMPOUND TURBINE-GENERATOR

inch vacuum, the best heat rate is expected to be about 10 500 B.t.u. per kw.-hr. for the turbine, generator, and auxiliaries.¹

208. Special Turbines. *Exhaust turbines, back-pressure turbines, and mixed-pressure turbines* are special forms of turbine employed occasionally where special conditions obtain. ***Exhaust-steam turbines*** receive steam exhausted by a steam engine or other source at about atmospheric pressure, and generate power as the steam passes through its several stages, finally being discharged to a condenser in which a high vacuum is maintained. The turbine is able to utilize very efficiently this low-pressure steam even though its volume becomes enormously large at the condenser pressure because adequate area of passage through nozzles and buckets can be provided. The exhaust-steam turbine is generally employed where there are several non-condensing engines as in older plants, such as steel mills and collieries. The exhaust steam from these engines would otherwise be discharged into the atmosphere and wasted. Exhaust-steam turbines, in design and appearance, are practically the same as regular high-pressure turbines except that all parts and passage areas are much larger, to care for the large volumes of steam. It is generally necessary to use a steam accumulator if exhaust steam is supplied intermittently through a cycle of a few minutes' duration, as with rolling-mill engines, steam hammers, and hoisting engines.

The steam accumulator is a large cylindrical vessel partially filled with water into which the exhaust steam is discharged at a pressure of 2 to 5 pounds above atmospheric. When there is an excess of exhaust steam, the pressure rises in the accumulator, some of the steam is condensed, and the latent heat is stored in the water. During short intervals of time, when the exhaust-steam supply is less than the demand of the turbine, the pressure falls in the accumulator and the stored heat is made available by the evaporation of some of the hot water. The functioning of the accumulator depends upon a small cyclic pressure variation.

Back-pressure turbines are turbines designed to discharge their steam at pressures above that of the atmosphere. In many

¹ See Power Plant Engineering, Nov. 1, 1929, pp. 1171-78, for more complete data and heat-cycle diagram.

plants, exhaust steam, at pressures varying from one or two pounds to 15 or 20 pounds per sq. in. gage, is employed for heating buildings or in carrying on industrial processes. In many large office buildings where power is required for lights and elevators, it is more economical to use high-pressure steam in a back-pressure turbine than to operate the turbine condensing and then employ live high-pressure steam for heating. By employing the turbine as a reducing valve the power is, in a way, a by-product obtained for a disproportionately small additional expenditure for fuel.

In many cases the heating load has a wide seasonal variation and the back-pressure turbine would have to be operated uneconomically for long periods during the non-heating periods. In that case, it is most profitable to employ an *extraction turbine*, an ordinary multistage condensing turbine, provided with hand-valves and piping for extracting steam from some intermediate stage in which the steam pressure is suitable for the heating load. During off-heating periods, all the steam passes through the entire turbine and into the condenser, thus producing power most economically.

A *mixed-pressure turbine* may be described as a back-pressure turbine to which an exhaust-steam turbine has been added. The high-pressure end receives steam direct from the boiler to help the low-pressure turbine over the periods of inadequate exhaust-steam supply. There are two extreme conditions of operation: (1) sufficient exhaust-steam available for the power demand from the turbine, and no high-pressure steam required, the high-pressure end of the turbine running idle; (2) no exhaust-steam available, and the turbine then being operated altogether by high-pressure steam.

209. Performance Data of Steam Turbines. Representative performance data and results for turbines varying in rated capacity from 10 kw. to 208 000 kw. are presented in Table 61 and Fig. 206. Reference to the sources of this data is indicated by letters which are explained just below the table. The private data is from sources considered to be reliable.

The heat rates (B.t.u. supplied at the throttle above the temperature of the feed water entering the boiler per kw. at the

switchboard) are given for the combined turbo-generator unit at the load giving the best economy. For all the larger units this load is about 75 or 80 per cent of the rated load. The full-

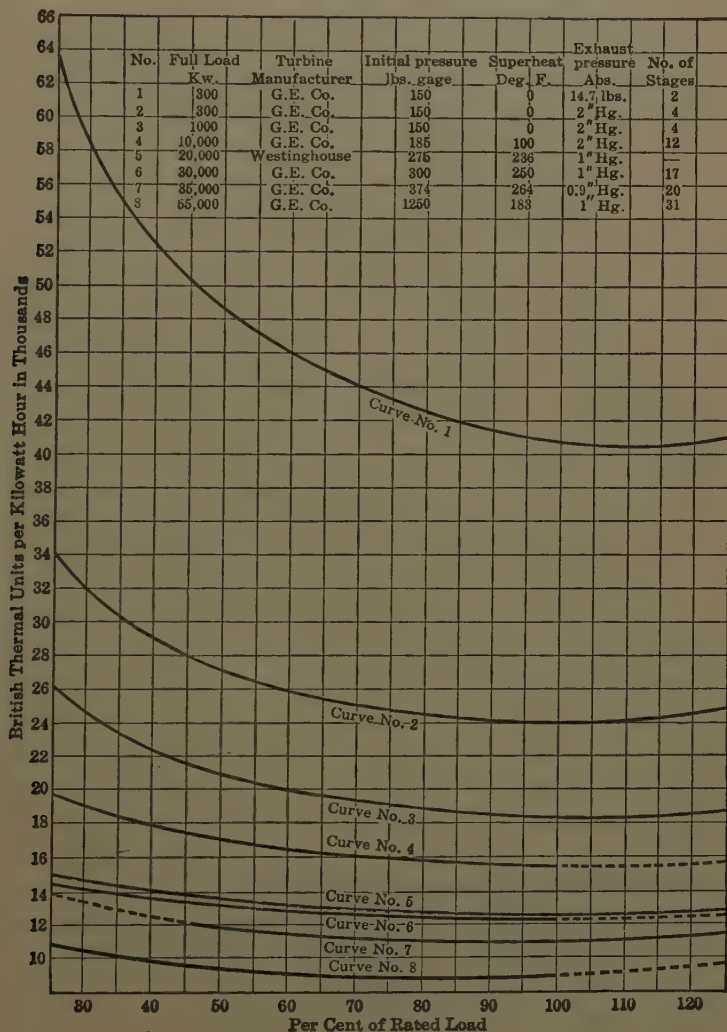


FIG. 206.—TURBINE HEAT-RATE CURVES

load efficiencies of electric generators vary from about 85 per cent for the smaller units to about 97 per cent for the largest units.

No performance results for reaction turbines with exhaust at

TABLE 61

REPRESENTATIVE PERFORMANCE RESULTS OF STEAM TURBINES

NUMBER AND REFER- ENCE †	RATED CA- PACITY KW.	R.P.M.	INITIAL PRES- SURE LBS. PER SQ. IN. ABS.	INITIAL SUPER- HEAT DEG. F.	EXHAUST PRES- SURE LBS. PER SQ. IN. ABS.	HEAT RATE B.T.U. PER KW. HR.	THER- MAL EF- FICIEN- CY PER CENT	NO. OF EX- TRAC- TION PTS.
WESTINGHOUSE TURBINE—GENERATORS								
1 (M)	1 000	3600	265	150	0.736	16 500	20.6	—
2 (P)	2 000	3600	265	150	0.736	15 900	21.4	—
3 (P)	5 000		265	150	0.736	15 500	22.0	—
4 (M)	10 000	1800	265	150	0.736	14 525	23.5	—
5 (G)	15 000	1800	215	125	0.785	15 300	22.3	—
6 (M)	15 000	1800	415	150	0.736	13 500	25.2	—
7 (N)	20 000	1800	281	196	0.785	14 700	23.3	—
8 (N)	35 000	1800	330	253	0.388	12 850	26.6	—
9 (A)	60 000	1500	220	150	0.491	13 950	24.5	—
10 (N ₂)	60 000	1800	305	281	0.407	11 247	30.3	2
GENERAL ELECTRIC TURBINE—GENERATORS								
11 (P)	10	3600	165	0	14.7	67 000	5.1	—
12 (P)	50	3600	165	0	14.7	52 400	6.5	—
13 (P)	100	3000	165	0	14.7	47 200	7.2	—
14 (P)	400	3600	165	0	14.7	36 800	9.3	—
15 (P)	100	3600	165	0	0.982	25 100	13.6	—
16 (P)	500	3600	165	0	0.982	20 150	16.9	—
17 (P)	1 000	3600	165	0	0.982	18 000	18.9	—
18 (P)	1 000	3600	165	100	0.982	15 530	22.0	—
19 (E ₁)	2 500	3600	290	150	0.736	14 340	23.6	—
20 (P)	5 000	3600	165	150	0.491	15 000	22.7	—
21 (P)	10 000	1500	195	100	1.30	15 010	22.7	—
22 (N)	20 000	1875	200	118	0.736	14 720	23.1	—
23 (N)	30 000	1800	315	250	0.736	12 720	26.8	—
24 (U)	35 000	1500	376	263	0.490	11 950	28.5	—
25 (U)	35 000	1500	374	264	0.462	10 950	31.1	3
26 (E ₂)	45 000		593	237	0.426	12 495	27.3	3
27 (N ₃)	55 000	3600—1800	1265	178	0.491	9 235	36.8	4
28 (E ₃)	75 000	1800	330	299	0.491	10 800	31.6	2
29 (E ₄)	208 000	1800	665	233	0.491	9 850	34.6	5

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30 (N)	1 250	3600	200	93	1.05	17 000	20.1	—
31 (E ₃)	1 500	3600	267	134	0.24	14 500	23.5	—
32 (N)	4 000	3600	211	119	0.383	15 800	21.6	—
33 (N)	15 000	1800	272	211	0.731	13 530	25.2	—
34 (N ₁)	35 000		378	247	0.620	12 610	27.0	2
35 (N ₂)	50 000	1800	601	190	0.575	10 680	31.9	4

† The reference letters indicate the source of the data as follows:

(A) Transactions A.S.M.E., 1922, p. 377.

(E₁) Power Plant Engineering, Feb. 1, 1929, p. 177.

(E₂) Power Plant Engineering, Feb. 1, 1927, p. 181.

(E₃) Power Plant Engineering, Sept. 1, 1929, p. 950.

(E₄) Power Plant Engineering, Nov. 1, 1929, p. 1174.

(E) Power Plant Engineering, Feb. 15, 1923, p. 227.

(G) Gebhardt, Steam Power Plant Engineering, 5th edition, p. 490.

(M) Data supplied by the manufacturer.

(N) Proceedings Natl. Electric Light Assn., vol. 83, 1926, pp. 1045-1096.

(N₁) Proceedings Natl. Electric Light Assn., vol. 84, 1927, pp. 1173-1188.

(N₂) Publication No. 289-98, Natl. Electric Light Assn., 1929.

(N₃) Publication No. 289-73, Natl. Electric Light Assn., 1929.

(P) Private data.

(U) Data furnished by the utility company operating the unit.

atmospheric pressure are included in Table 61. Turbines larger than 300 or 400 kw. are seldom built for atmospheric exhaust, and turbines for non-condensing operation are nearly always of the impulse type.

Turbine results numbered 24 and 25 were obtained, by test, in 1925, on one and the same turbine which is installed in the Long Beach Plant No. 2 of the Southern California Edison Company. This turbine is shown in Fig. 202. The results in line 24 were obtained without steam extraction for feed-water heating, and the test data recorded in line 25 was obtained with steam extraction from the 11th, 15th, and 18th stages in accordance with the turbine design.

Because of the difficulties involved in segregating and weighing the various quantities of steam leaving an extraction turbine, comparatively few test data are available at this time (March, 1930) for turbines operating on the regenerative cycle. The guarantees as to steam consumption and thermal efficiency under which most regenerative turbines have been sold by manufacturers have so far been based upon turbine performance determined without extraction. Hence the results in lines 24 and 25 are of particular interest. The results in line 26 represent the average monthly performance of two 45 000-kw., two-

cylinder, tandem 26-stage General Electric turbines at the Columbia Power Company's plant located near Cincinnati, Ohio. The turbines are operated with reheat and steam extraction from the 18th, 22nd, and 24th stages. The overall thermal efficiency of this plant from coal pile to station send-out is 27.3 per cent, which was a world's record when it was attained in December, 1926.

It will be noticed that the heat rates in Table 61 vary from 67 000 B.t.u. per kw. per hr. for the smallest turbo-generator unit to approximately 9200 B.t.u. for the most efficient.

It should be remembered that the performance results given in Table 61 are for new turbines of each size and type. For the extraction turbines the heat rates are correct only for the precise heat cycle for which the turbine was designed, or on which the tests were carried out. The results in many instances were secured under rigid test conditions which gave the best possible performances. As the turbines age, and under operating conditions prevailing in many power plants, the average daily performances are from 5 to 25 per cent poorer than those stated in the table.¹

Although the data in Table 61 and the curves in Fig. 206 do not indicate it, the aim of designers of large modern turbines is to so proportion the turbine wheels and buckets as to give the minimum heat consumption at about 75 or 80 per cent of the full-load rating of the turbine.

210. Turbine Performance at Fractional Loads. Heat-rate curves for turbines varying in rated capacity from 300 kw. to 55,000 kw. are shown in Fig. 206. These curves are for the combined turbine and generator unit. It can be seen from these curves that the heat rate at 25 per cent load varies from about 1.6 times the full-load rate for the 300-kw. non-condensing machine to about 1.15 times full-load rate for the 55 000 kw. machine (see Table 62).

It is interesting to compare these figures with corresponding

¹ S. S. Sanford, *Effect of Use upon the Steam Rate of Turbine-Generators*, Power, Mar. 23, 1926, p. 436.

TABLE 62
FACTORS FOR COMPUTING FRACTIONAL LOAD HEAT RATES

CURVE No.	RATED KW.	EXHAUST PRESSURE	FACTORS			TYPE OF ENGINE
			25%	50%	75%	

STEAM TURBINES (FIG. 206)

1	300	Atmos.	1.58	1.19	1.07	
2	300	2" Hg. abs.	1.42	1.13	1.04	
3	1 000	2" Hg. abs.	1.42	1.13	1.05	
4	10 000	2" Hg. abs.	1.28	1.09	1.03	
5	20 000	2" Hg. abs.	1.20	1.08	1.01	
6	30 000	2" Hg. abs.	1.18	1.10	1.02	
7	55 000	1" Hg. abs.	1.15	1.02	0.99	

STEAM ENGINES (FIG. 145)

	I.hp.					
1	30	Atmos.	1.11	1.03	1.01	Simple
2	80	Atmos.	1.26	1.08	1.02	Simple
3	350	Atmos.	1.49	1.06	0.99	Simple
4	900	Atmos.	1.41	1.03	0.98	Simple
5	170	Atmos.	1.06	1.003	0.99	UnafLOW
6	21" × 22"	Atmos.	1.004	0.99	0.98	UnafLOW
7	300	Cond.	1.47	1.07	0.94	Comp.
8	1,500	Cond.	1.46	1.11	0.99	Comp.
9	430	Cond.	1.08	1.00	0.97	UnafLOW
10	5,500	Cond.	1.22	1.09	1.02	Triple exp.
11	1,500	Cond.	1.12	1.03	0.98	UnafLOW

ones for steam engines computed from the curves of Fig. 145, and incorporated in Table 62. The performance of all unafLOW engines are especially noteworthy in this respect.

The fractional-load steam rates for a certain line of small single-stage non-condensing impulse turbines are found by multiplying the full-load steam-rates by the fac-

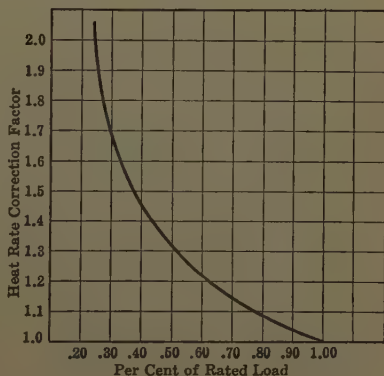


FIG. 207.—HEAT-RATE CORRECTION FACTORS

tors indicated by the curve of Fig. 207. This line of turbines varies from 5 to 350 rated horsepower. For 2, 3, or 4-stage machines, different curves would be required to suit the design speed, and size range of the turbines under consideration.

211. Steam-Rate and Heat-Rate Corrections. Turbines are generally sold with a guarantee by the manufacturer that under certain specified conditions of load, initial steam pressure, initial superheat, and exhaust pressure the heat rate will not exceed certain values. When the acceptance tests are conducted on the unit, it may be impossible to maintain one or more of the conditions specified in the purchase contract. It is then necessary to apply suitable corrections in order to reduce the test results to the basis specified in the contract. Usually these corrections should be agreed upon when the contract is signed or it should be agreed to test the unit under varying conditions which will provide the data from which correction curves may be prepared. These corrections vary to some extent for different units, depending upon the type, size, speed, number of stages, initial pressure,

initial superheat, etc.

For a certain 30 000-kw. 17-stage 1800-r.p.m. turbine-generator designed by the General Electric Company to operate with an initial steam pressure of 300 lbs. per sq. in. gage and 250° F. of superheat, the pressure correction used was 0.2 per cent correction in steam rate for each 10 lbs. pressure variation above or below

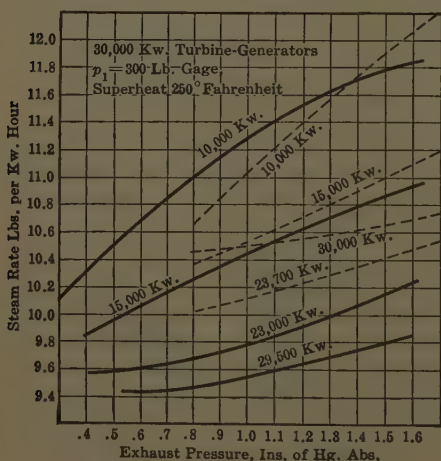


FIG. 208.—VACUUM CORRECTIONS

300 lbs. gage, the range of pressure over which this correction is to be applied being from 285 to 315 lbs. The steam rate is corrected 1 per cent for each 11° F. above or below the 250° of super-

heat considered normal for this turbine. Actual tests of the turbine were carried out from which corrections for changes in vacuum at different loads were determined as shown by the full-line curves of Fig. 208.

A certain 30 000-kw. 1800-r.p.m. machine was designed and built by the Westinghouse Company to operate normally with an initial pressure of 300 lbs. per sq. in. gage, and 250° F. of superheat. The pressure correction for steam rate is 1 per cent for each 25 lbs. above or below the standard of 300 lbs. This correction applies only between 275 and 325 lbs. The superheat correction used is 1 per cent for each 20° F. above or below the standard of 250 degrees. This correction applies between superheats of 225° F. and 300° F. Steam-rate changes for different vacuums and various loads on the Westinghouse turbine are shown in Fig. 208 by the dash-line curves.¹

For small turbines with ratings up to 3000 kw., and for either condensing or non-condensing operation, the improvement in steam rate is about one per cent for increments of superheat

TABLE 63

FULL-LOAD STEAM-RATE CORRECTIONS FOR PRESSURE CHANGES
(Turbine r.p.m. in All Instances 3600)

No.	RATED POWER	No. STAGES AND TYPE	EXHAUST PRESSURE	STEAM RATES AND CORRECTIONS AT VARIOUS INITIAL PRESSURES					
				Initial Pressure	200	175	150	125	100
1	60 hp.	1 Re-entry Impulse	Atmos.	Steam Rate per Cent Change	35.4 86.5 -13.5	38.4 93.6 - 6.4	41.0 100.0 0.0	44.2 107.8 + 7.8	48.0 117.0 +17.0
2	160 hp.	1 Re-entry Impulse	Atmos.	Steam Rate per Cent Change	30.3 87.5 -12.5	32.3 93.4 - 6.6	34.6 100.0 0.0	37.2 107.2 + 7.2	40.3 116.2 +16.2
3	100 kw.	3 Impulse	2" Hg. Abs.	Steam Rate per Cent Change	23.6 95.5 - 4.5	24.2 98.0 - 2.0	24.7 100.0 0.0	25.7 103.2 + 7.2	26.7 116.2 +16.2
4	300 kw.	4 Impulse	2" Hg. Abs.	Steam Rate per Cent Change	20.3 95.4 - 4.6	20.8 97.6 - 2.4	21.3 100.0 0.0	22.0 103.2 + 3.2	22.4 105.2 + 5.2
5	2500 kw.	4 Impulse	2" Hg. Abs.	Steam Rate per Cent Change	15.4 95.1 - 4.9	15.8 97.6 - 2.4	16.2 100.0 0.0	16.7 103.0 + 3.0	17.4 107.3 + 7.3

¹ The steam-rate correction data for these two turbines was taken from Proceedings, National Electric Light Assn., vol. 83, 1926, pp. 1051 and 1059.

varying in value from 10° to 15° F. each, between zero and 200° of superheat.

Full-load steam-rate corrections for various initial steam pressures between 100 and 200 lbs. per sq. in. gage are shown in Table 63. The corrections for the 60 and 160 hp. non-condensing turbines was computed from the curves of Fig. 209. In comparing the steam rates on a percentage basis, the steam rate for 150 lbs. is taken as 100 per cent throughout the table. It will be noted that for the range of sizes given in Table 63, the rated capacity of the turbine has but little effect on the steam-rate corrections for pressure changes. However, there is considerable

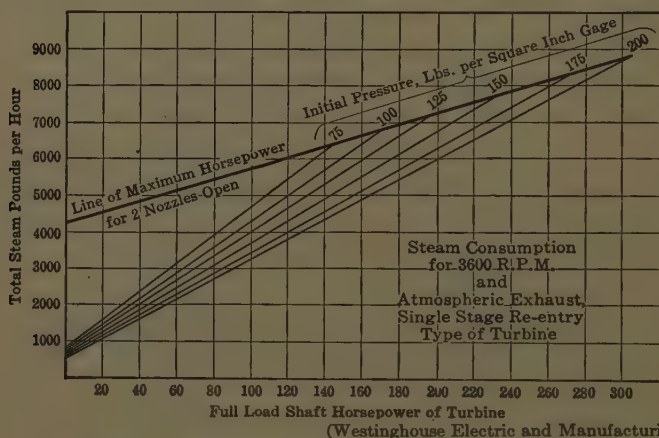


FIG. 209.—WILLANS LINES FOR VARIOUS PRESSURES
(Westinghouse Electric and Manufacturing Co.)

difference in the pressure corrections for small turbines designed for non-condensing and condensing operation. There is also considerable difference in the value of the corrections for turbines of large differences in capacity, as will be seen by comparing the corrections for the turbines of Tables 63 and 64 and the 30 000-kw. turbines for which data is given in the paragraphs above. There is also a tendency for the corrections to decrease as the initial pressure rises. This is largely due to the fact that at higher pressure, the number of B.t.u. made available by the adiabatic expansion accompanying a 25-lb pressure drop, as read from the Mollier diagram, progressively decreases with rising pressure throughout the entire range up to the critical pressure.

The data in Table 64 is for a turbine designed to operate with an initial steam pressure of 200 lbs. per sq. in. gage, initial super-heat 100° F., and exhaust pressure 2 in. of mercury absolute. It was considered that the full-load steam-rate correction to be applied was one per cent for each 10 lbs. change in pressure above or below the base pressure of 200 lbs. per sq. in. gage.

TABLE 64

CHANGE IN STEAM RATE FOR 10 000-KW. 12-STAGE 1500-R.P.M. TURBINE

LOAD IN KW.	2500	5000	7500	10,000
S.R. at I.p. 200 lbs. gage.....	16.5	14.0	13.05	12.6
S.R. at I.p. 185 lbs. gage.....	17.45	14.25	13.25	12.8
Change in S.R. per cent.....	+ 5.8	+ 1.8	+ 1.5	+ 1.6

212. Weights of Steam Turbines. The approximate weights of turbines of various rated capacities are given in Table 65. Unit No. 9 is a General Electric 1250 lb. pressure turbine. All others are Westinghouse Machines.

TABLE 65

WEIGHTS OF STEAM TURBINES WITHOUT GENERATOR

NO.	RATED POWER	R.P.M.	EXHAUST PRESSURE	TOTAL WT. LBS.	UNIT WT. LBS. PER KW.	UNIT WT. LBS. PER HP.
1	5 hp.	3600	Atmos.	500		100
2	50 hp.	3600	Atmos.	500-800		10 -16
3	100 hp.	3600	Atmos.	800-1800		8 -18
4	400 hp.	3600	Atmos.	1800-2600		4.5- 6.5
5	500 kw.	3600	Cond.	19 000	38	
6	1 000 kw.	3600	Cond.	23 000	23	
7	10 000 kw.	1800	Cond.	185 000	18.5	
8	25 000 kw.	1800	Cond.	500 000	20.0	
9	55 000 kw.	3600/ 1800	Cond.	925 000	16.8 *	
10	62 500 kw.	1200	Cond.	1 000 000	16.0	
11	80 000 kw.	1800	Cond.	2 000 000	25.0 *	
12	110 000 kw.	1800	Cond.	2 120 000	19.3 *	

* These weights include both turbine and generator.

The weight of any machine always furnishes an approximate index as to its cost. All machines are constructed of cast iron,

various kinds of steel, and small quantities of bronzes, copper, or other metals whose prices, in the crude state, are quoted regularly in the trade journals. Then each machine requires the expenditure of a certain amount of labor per unit of weight. The selling price of most large machines like engines, turbines, generators, and pumps will be found to vary between 20 cents per lb. for large centrifugal pumps requiring comparatively little machining and little labor in assembling, to perhaps \$2.00 per lb. for small steam turbines requiring much machine work and considerable labor to assemble the many parts.

PROBLEMS

1. Supply all transformation steps which have been omitted in deriving equation (432), p. 412, from (426).

2. If steam at an initial pressure of 500 lbs. per sq. in. gage and a temperature of 670° F. expands adiabatically through a nozzle until the quality equals 0.80: (a) What is the final pressure? (b) The theoretical velocity of the steam leaving the nozzle? (c) What is the value of the kinetic energy developed in ft.-lbs. per lb. of steam per second? (d) If the throat diameter of the nozzle having a circular cross-section is 0.95 in., compute the weight of steam flowing in lbs. per sec. by Grashof's formula.

3. Compute the lbs. of steam per sec. flowing from a nozzle for the following conditions: $P_1 = 150$ lbs. per sq. in. gage, $t_1 = 458.5^\circ$ F., $P_2 = 110$ lbs. per sq. in. abs., throat diam. = 1.25 in., circular cross-section. What is the exit velocity if the steam expansion is adiabatic?

4. Compute the Carnot and Rankine-cycle efficiencies for the conditions (a) in Problem 2, (b) in Problem 3.

5. Compute the Carnot and Rankine-cycle efficiencies for turbine No. 25, Table 61. What would be the heat rates of C. and R. ideal engines operating under these conditions? Repeat for turbine No. 27.

6. Compute the nozzle-and-bucket efficiency of a single-stage impulse wheel carrying a single-bucket row when $p_1 = 125$ lbs. gage, $t_1 = 460^\circ$ F., $P_2 = 14.7$ lbs. abs., nozzle angle = 18° , nozzle-velocity coefficient = 0.97, bucket inlet angle = 22° , exit angle = 20° , pitch diam. of wheel = 30 in., r.p.m. = 3600. (Ten inch slide-rule work is not sufficiently accurate).

7. Determine, by use of graphical methods, suitable bucket angles for a double-bucket row wheel and intermediates for the following conditions: nozzle-exit angle = 20° , nozzle-velocity coefficient = 0.97, $P_1 = 150$ lbs. per sq. in. gage, $t_1 = 416^\circ$ F., $p_2 = 15$ lbs. per sq. in. abs., pitch diameter of wheel = 36 in., and r.p.m. = 3600.

8. Compute the nozzle-and-bucket efficiency for an impulse wheel carrying a double-bucket row, for the conditions as found in Problem 7.

9. A single-stage impulse turbine is to be built for the following specifications: pitch diameter of wheel = 30 in., r.p.m. = 3600, two rows of moving

buckets, nozzle angle = 18° , nozzle-velocity coefficient = 0.965, inlet and exit angles have the following values:

	<i>Inlet</i>	<i>Outlet</i>
1st row moving buckets	20°	20°
Intermediates	27°	20°
2d row moving buckets	30°	25°

mean height of moving buckets, $\frac{h_1 + h_2}{2} = 0.128$ ft., at full load 10 nozzles are open, nozzle-throat diameter = 0.59 in., nozzle-mouth height = 0.80 in., ten nozzles cover 16 per cent of the wheel periphery at the pitch line. Consider bearing-friction loss constant at all loads and equal to 1.0 per cent of the full-load horsepower, initial steam pressure $p_1 = 135$ lbs. gage, $t_1 = 468^\circ$ F., $p_2 = 15$ lbs. abs.

Compute items (1) to (7) for full load (10 nozzles open):

- (1) weight of steam flowing, lbs. per sec. and lbs. per hr.,
- (2) nozzle-and-bucket efficiency,
- (3) rotation loss, as a decimal fraction of the energy supplied,
- (4) wheel efficiency,
- (5) wheel horsepower,
- (6) shaft horsepower,
- (7) steam rate per shaft horsepower per hr.,
- (8) repeat (1) to (7), inclusive, when only 3 nozzles are open,
- (9) from the above results, draw a Willans line on engineers' cross-section paper. From the Willans line compute and plot a steam-rate curve,
- (10) compute the full-load thermal efficiency referred to the S.h.p.,
- (11) compute the full-load Carnot and Rankine ideal efficiencies,
- (12) compute the Rankine-cycle efficiency ratio.

10. Check the full-load steam rate computed in Problem 9 by means of Parsons' approximate method.

11. The 10 000-kw. Curtis turbine No. 21 in Table 61 contains 12 stages. The first wheel carries two rows of buckets, all the remaining wheels carry a single-bucket row. The pitch diameter of the first-stage wheel = 84 in. The diameter of the wheels from the second to the eleventh stages inclusive is 73 in. each. The twelfth-stage wheel diameter = 87.5 in. Compute the full-load steam rate and heat rate by the Parsons' method, and compare with the result in Table 61.

12. Find the equation for the Willans line of Problem 9.

13. Specifications for a 2-stage impulse turbine are as follows: rotative speed = 3000 r.p.m., pitch diameter for both wheels = 36.0 in., each wheel carries a double-bucket row, initial pressure 150 lbs. per sq. in. abs., $t_1 = 408^\circ$ F., exhaust pressure = 2 in. Hg. abs., nozzle angle = 20° , nozzle-velocity coefficient = 0.96 for both stages, first-stage nozzles are round and the throat diameter = 0.60, number of first-stage nozzles = 18, which cover 18 per cent of the wheel periphery. The second-stage nozzles are round also, and have a

throat diameter of 0.85 in. covering 70 per cent of the wheel periphery. The bucket heights are

	<i>1st row</i>	<i>2d row</i>
First stage	1.00"	1.54"
Second stage	1.55"	2.75"

At full load, all nozzles in both stages are open. At fractional loads, part of the first-stage nozzles are closed but all second-stage nozzles remain open.

Make the bucket exit angles about 15 degrees less than the inlet angles, but no exit angles are to be less than 20 degrees. Assume the second-stage flow to be 2 per cent less than the first due to leakage. Assume bearing friction constant at 1.0 per cent of full-load shaft horsepower.

Compute, for full-load conditions:

- (1) the steam pressure entering the second-stage nozzles,
- (2) determine graphically the correct nozzle angles,
- (3) total steam flow, lbs. per sec.,
- (4) nozzle-and-bucket efficiency for each stage,
- (5) rotation loss (as a decimal) for each stage,
- (6) wheel efficiency for each stage,
- (7) total shaft horsepower,
- (8) steam rate per shaft horsepower per hour,
for fractional load (only 4 first-stage nozzles open)
- (9) repeat computations for items (1) to (8) inclusive,
- (10) from the above two points plot a Willans line; and compute and draw a steam-rate curve for loads varying from 1/4 to 4/4 load,
- (11) compute Carnot and Rankine-cycle efficiencies,
- (12) compute actual thermal efficiency at full load, and the Rankine-cycle efficiency ratio.

14. A 50 000-kw. regenerative turbine operates under the following conditions: Initial steam pressure $p_1 = 500$ lbs. per sq. in. abs., initial superheat = 150° F., exhaust pressure = 1 in. Hg. abs., steam for heating feed water is extracted at three pressures, 5 lbs. abs., 25 lbs. abs., and 80 lbs. abs. Assume the ideal condition that the feed water is raised to the temperature of saturated steam at the pressure in the respective heaters. Draw to scale the temperature-entropy diagram showing this ideal regenerative cycle. Determine the thermal efficiency of this cycle by evaluating the proper areas on the diagram. Compute the corresponding Carnot and Rankine-cycle efficiencies, and compare the three efficiencies thus found.

15. A certain large turbine operates on the combined reheat-regenerative cycle. The conditions of operation are as follows: $p_1 = 1215$ lbs. abs., initial temperature of steam = 750° F. The steam expands adiabatically to a pressure of 400 lbs. abs. It is then returned to a reheat boiler. From there it returns to the turbine at 385 lbs. abs. and temperature of 725° F. Steam is extracted for feed-water heating at three points, the pressures being 5 lbs. abs., 45 lbs. abs., and 140 lbs. abs., Assume, as in Problem 14, that the feedwater leaves each heater at the temperature of the steam entering the respective heaters.

Draw to scale the TN -diagram for these conditions. From the appropriate areas on this diagram compute the thermal efficiency of this combined cycle. Compute the corresponding Carnot and Rankine-cycle efficiencies. (The necessary values for plotting may be obtained from the Mollier diagram in *Mechanical Engineering*, February, 1929, or the *Steam Tables and Mollier Diagram* by J. H. Keenan.

16. Make a study to determine whether the wheel efficiency of the two-stage turbine of Problem 13 could be improved by changing the pressure of the steam entering the second-stage nozzles. (This pressure may be changed by changing the number or size of the second-stage nozzles. If the throat diameter of the nozzles is changed, the nozzle-mouth height and the length of buckets must be changed accordingly.)

17. Using data from the heat balance of Fig. 189*b*, compute the heat rate for the data given.

18. Supply the missing steps between (431) and (437) to show that the maximum value of S_0 is equal to the velocity of sound in the medium flowing.

CHAPTER XVI

INTERNAL-COMBUSTION ENGINES

213. **Brief Historical Notes.**¹ *Abbé Hautefeuille*, the son of a baker at Orleans, seems to have been the first to propose, in 1678, an explosion motor which employed gunpowder fuel for pumping water. This was more than three hundred years after the first recorded use of gunpowder at the battle of Cressy in 1346. The gunpowder was to be burned in a separate chamber which communicated with the water reservoir. The expanded gases of combustion largely escaped into the atmosphere through a valve. As the remaining gases cooled, a partial vacuum was produced in the chamber into which the water was forced by atmospheric pressure. There is no evidence that Hautefeuille ever built an operable engine.

In 1680 *Christian Huygens*,² (1629–1695) Dutch philosopher and mathematician, constructed and exhibited a working model of an engine comprising a cylinder and piston. The explosion of a small quantity of gunpowder expelled the air through collapsible leather tubes which served as check valves. Thus a partial vacuum was produced in the cylinder and the atmospheric pressure drove the piston into the cylinder doing useful work. This was probably the model upon which Newcomen's and Watt's atmospheric engines were patterned. Papin attempted to improve upon this engine by perfecting the valves. But he found that, in spite of his best efforts, nearly one-fifth of the air remained in the cylinder to interfere with the full movement of the piston. Thereupon he abandoned the explosive engine and directed his efforts to developing the steam engine, which he considered to be more practicable.

Nothing more seems to have been done to develop the explosive engine for more than a hundred years. On May 7, 1794, *Robert*

¹ See Byran Donkin, *Gas, Oil, and Air Engines*; Encyclopaedia Britannica.

W. L. Lind, *Internal Combustion Engines*.

² For short biographical sketch see Engineering, Apr. 12, 1929, p. 452.

Street was granted an English patent on an engine which was a long step forward. It consisted of a vertical cylinder with a piston. The piston was connected to a lever which operated a pump. The bottom of the cylinder was heated by an external fire. A few drops of turpentine were introduced into the cylinder and evaporated by the heated cylinder head. The piston was then drawn up through a portion of the stroke taking in air which formed an explosive mixture with the turpentine vapor. Ignition was secured by an open flame projected through a port uncovered by the piston. The resulting explosion drove the piston to the end of the stroke forcing the pump piston down, thus performing useful work. There was no compression and the mechanical details were crude, but the unit contained several of the essential elements of a successful engine.

Philippe Lebon, a French engineer, was granted two patents, one in 1799 and the other in 1801. The first related to the production of gas by distillation of coal. In the second, he proposed to utilize the gas to operate a piston in a cylinder very similar to the steam engine. The inflammable gas and the air were to be introduced separately into the cylinder, compressed to a slight extent, and then ignited by an electric spark. This engine was double-acting. So far as known, *Lebon was the first to propose the use of the electric spark for ignition*. He also seems to have appreciated the importance of compressing the charge before ignition. Unfortunately before he had been able to reduce his invention to practice he was assassinated in 1804.

The honor of having been the *first to employ compression* in the internal-combustion-engine cycle belongs to *William Barnett*, who was granted three English patents in 1838 which represented another decided advance.

The mechanical details of Barnett's third engine differed but little from modern engines. The air and coal gas were compressed in separate cylinders before being introduced into the main cylinder. The engine was double-acting, a charge exploding alternately on each face of the piston. The burned gases were expelled during the first half of the back stroke through a port in the center of the cylinder. This port was closed by the piston,

The gas and air, somewhat compressed in the auxilliary cylinders, were then introduced into the main cylinder and further compressed together with the residual burned gases from the previous explosion as the piston continued to the end of its stroke. The ignition was produced as the piston reached the end of its stroke by a flame enclosed in a rotating cage in which a port registered at the proper instant with a port in the cylinder head. The explosion usually extinguished the ignition flame. An auxiliary gas flame burning continuously just outside the rotating cage relighted the ignition flame before the next explosion occurred. This ignition device was very good, and it continued in use on many engines as late as 1890. Barnett built and operated several engines. Many details were imperfectly developed, and his engines never passed the experimental stage.

In 1860 *Jean J. E. Lenoir*,¹ a Frenchman, put on sale regularly to the public for the first time in history a commercial internal-combustion engine. Between 1860 and 1865 nearly 500 Lenoir engines varying in size from 1/2 horsepower to 20 horsepower each were sold. It was a double-acting engine with slide-valve, cross-head, and guides somewhat similar to Barnett's design. Ignition was secured by electric battery and jump spark. There was no compression. The gas and air in proper proportions were drawn into the cylinder by the piston during the first half of the stroke, then ignition occurred, and if the spark were retarded, as often happened, the piston had passed through the major part of its stroke when the explosion occurred, thus there was very little expansion. Because there was no compression, the engine was easy to start and its operation was smooth and silent. But the lack of compression made the fuel consumption much higher than it was at first reported to be. The first official test, in 1861, of a 1/2 horsepower engine at 130 r.p.m. using Paris artificial gas (gas to air ratio 1 : 10) showed a consumption of 112 cubic feet per horsepower per hour, or about 5 times the consumption of average modern engines. The cooling and lubricating facilities provided were very inadequate. For these reasons the engine sales practically ceased about 1865. However, the small success of this

¹ Granted U. S. Patent No. 31 722 Mar. 19, 1861.

engine was sufficient to demonstrate to the public the many advantages of the internal-combustion engine. They were easy to transport and could be set up and started without delay. They could be readily built in small sizes and used advantageously to replace manual labor. All of which furnished a strong incentive to inventors and engineers for further development.

In 1862 M. Beau de Rochas, a French engineer, was granted a patent on an internal-combustion engine operating on a cycle which embodied all the essential features of modern efficient engines. His proposed cycle of operations, in all respects, is in use to-day in the so-called Otto-cycle engine. He stated the following four conditions as being essential to high efficiency in a practical engine:

- (1) The largest possible cylinder volume with the smallest cooling surface.
- (2) Maximum piston speed.
- (3) Maximum pressure at the beginning of the piston stroke
- (4) The greatest possible expansion.

These principles, when first announced, were considered revolutionary in the study of I.C. engines. But time has proven them to be of fundamental importance. All of the four conditions are essential, but the third is by far the most important in realizing high efficiency *It can be fulfilled only by employing compression before ignition of the charge.*

It was proposed to carry out this cycle during 4 strokes of the piston on one single-acting cylinder:

- 1st stroke, drawing in the charge of gas and air,
- 2nd stroke, compression of the charge,
- 3rd stroke, ignition at beginning of stroke and expansion throughout the stroke,
- 4th stroke, expulsion of the spent gas to the atmosphere.

This cycle is known as the four-stroke cycle of Beau de Rochas. It is a theoretical, ideal cycle in the same sense as the Carnot. This will be demonstrated later. It is charged by the Germans that de Rochas never attempted to construct an engine, and never worked out a completed design for one. He did write a

pamphlet, and was granted a French patent. The pamphlet was not published and lay hidden in some library until 1886 when it came to light in patent litigation against Otto's engine. It was only at this time that the engineering world was first made acquainted with the invention or proposals of de Rochas.

Nicolaus A. Otto (1832–1891) and *Eugen Langen* (1833–1895) exhibited their first engine at the Paris World's Fair in 1867.

This was a free-piston engine without compression, Fig. 210,



(Otto Engine Co.)

FIG. 210.—FIRST COMMERCIALLY SUCCESSFUL ENGINE BY OTTO AND LANGEN.

which closely resembled the first free-piston engine of record proposed in 1857 by the Italians, Barsauti and Matteucci.¹

The piston, in a vertical cylinder, carried a rack which drove a gear wheel through a cleverly arranged clutch, the clutch engaging the gear only on the upward stroke of the piston. The piston made the return stroke by simply falling unconstrained to the bottom of the cylinder. This Otto and Langen engine was very noisy and vibrated excessively, due to impact and recoil. It was built only in small sizes ($\frac{1}{3}$ to 3 horsepower each).

This was the first commercially successful gas engine. It was a long step in advance of its immediate predecessor, the

Lenoir engine, as to general design, construction, operating characteristics, and fuel economy. Tests of engines at the

Vienna exhibition of 1873 having a

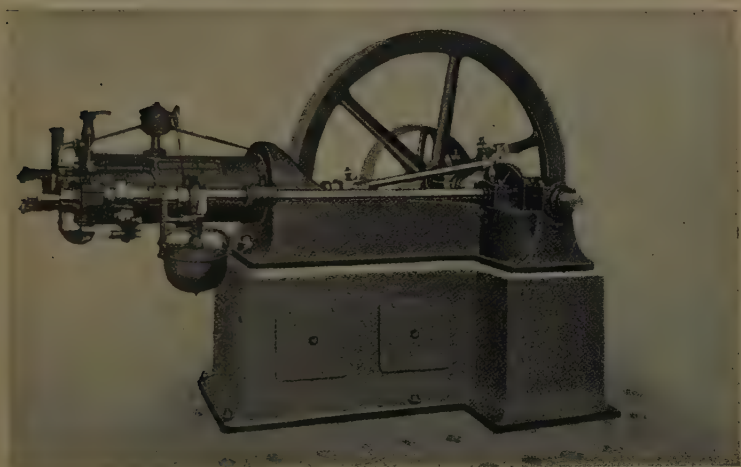
piston diameter of 8.75" with a stroke of 41.3" when running 80 to 85 r.p.m. developing 0.9 of a brake horsepower consumed coal gas at the rate of 40 cubic feet per brake horsepower per hour. This corresponds to a brake thermal efficiency of approximately 10 per cent.²

Meanwhile Otto was experimenting and studying to improve

¹ See Dugald Clerk, *The Gas and Oil Engine*, 2d edition, p. 10.

² See Carpenter and Diederichs, *Internal Combustion Engines*, p. 247.

and perfect both the cycle and the engine. Entirely independently of de Rochas, he re-invented the four-stroke cycle. First, drawing in by suction of the piston a charge of gas and air; second, compressing the charge; third, ignition and expansion during the power stroke; and fourth, clearing the cylinder of the burned gases. An engine of this type, Fig. 211, embodying many mechanical improvements was exhibited at the Centennial Exhibition in Philadelphia in 1876. This has come to be known as the Otto engine. Modern forms of this engine drive our motor cars and



(Otto Engine Co.)

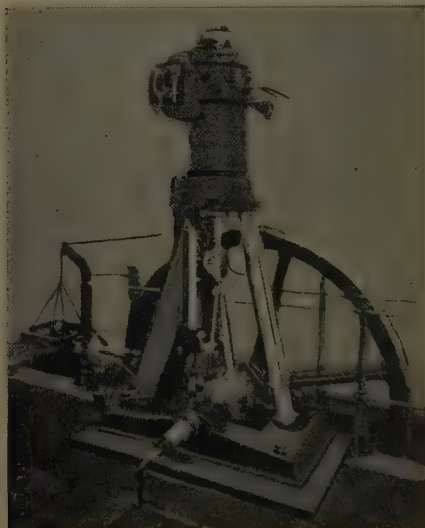
FIG. 211.—OTTO AND LANGEN ENGINE OF 1876

airplanes. All the small stationary low-compression gas or gasoline engines of commerce also operate on the Otto or de Rochas cycle. The advent of this Otto engine marked the real beginning of the successful application of internal-combustion engines to commercial work. To Beau de Rochas and to Otto belong jointly the honor of having invented the successful cycle. But to Otto alone belongs the credit for having recognized the merit of this cycle and for having the ingenuity and skill to embody the new principles in a practicable machine.

However, this engine, even though it is in use to-day fundamentally unchanged, is not all that could be desired. It has only

one working stroke in every four; three-fourths of the piston strokes absorb energy. Thus the unit weight of the engine is necessarily high, and the mechanical efficiency is low.

Otto and Langen held a basic patent, granted in 1877, which prevented other manufacturers from building four-stroke cycle engines up to about 1885. At that time, litigation over the patents of Otto were decided in Germany and England. The German courts held that the patents had been anticipated by Beau de Rochas. However, the English courts sustained the Otto patents. No suit was brought in the United States, but no doubt the European situation encouraged many infringements. The basic patent of Otto expired in 1894.



(Jour. A.S.M.E. June 1912)

FIG. 212.—DIESEL'S FIRST SUCCESSFUL ENGINE

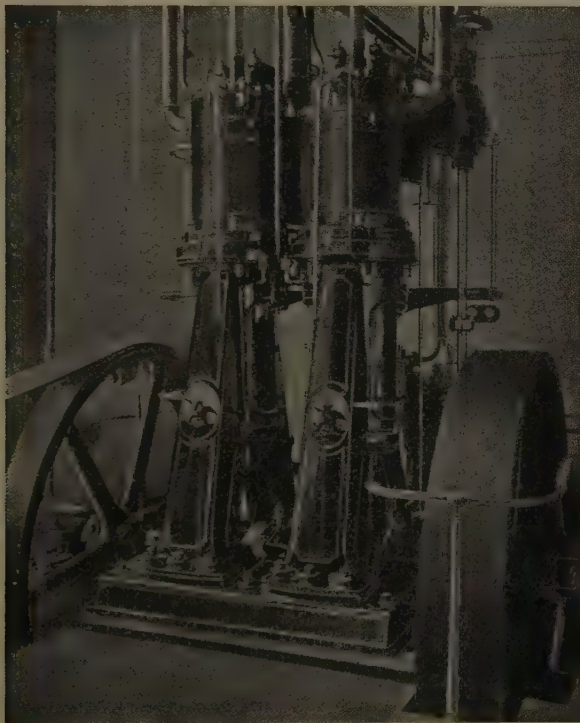
Up to the time of the German court decision, there had been strong incentive to improve the internal-combustion engine, and thus avoid infringement of the Otto patents. Among such efforts was one to develop an engine in which an explosion would occur every revolution. Dugald Clerk, an English inventor, was the first to succeed. The first model

of his two-cycle engine was first exhibited in 1880.¹ When the Otto patent was not sustained in Germany, many engine builders abandoned, for the time, the two-cycle engine because of its inherent defects, which included among others high fuel-consumption and lack of simplicity in its early forms.

In 1892, *Rudolph Diesel*, a German engineer, was granted patents on the constant-pressure high-compression cycle now

¹ See Dugald Clerk, *The Gas and Oil Engine*, 6th edition, p. 187.

known universally as the Diesel cycle.¹ The distinctive feature of this cycle is the compression of air, nearly adiabatically, to such a pressure that the resulting temperature is above the kindling point of the fuel. At the beginning of the power stroke, the liquid fuel is injected into the cylinder in a pulverized state. It ignites spontaneously in the highly compressed and highly heated air,



(Busch-Sulzer Bros. Diesel Engine Co.)

FIG. 213.—FIRST DIESEL BUILT IN AMERICA

burning as rapidly as it is injected. Still it is by no means an explosive combustion as is the case in the Otto engine. The injection usually begins 10 to 15 degrees of crank angle before the top dead center is reached, and it continues until the piston has completed 5 to 10 per cent of its forward or downward stroke.

¹See Rudolph Diesel, *The Present Status of the Diesel Engine in Europe, and a Few Reminiscences of the Pioneer Work in America*, Journal A.S.M.E., June, 1912, p. 509.

The cycle may be carried out in four strokes, or in two strokes, as in the Otto engine. Early models of Diesel's engines are shown in Figs. 212 and 213.

214. The Otto Ideal Four-Stroke Cycle. The events and their relations of the Otto four-stroke cycle are illustrated in Fig. 214. Starting with the intake stroke when the piston is in its extreme left-hand position, as it moves forward (to the right) the spring-loaded intake valve opens, and a fresh mixture of fuel and air is

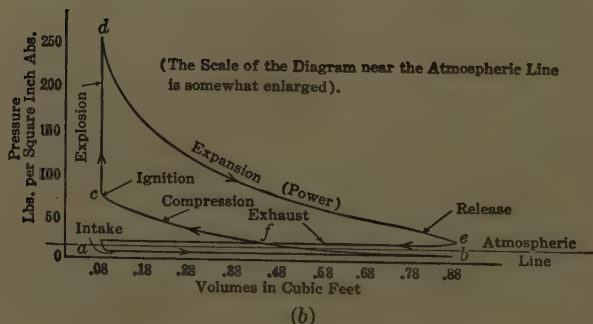
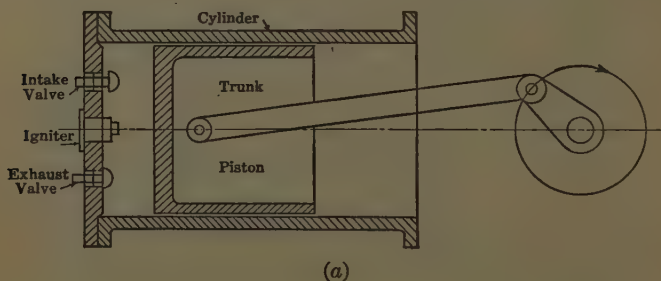


FIG. 214.—FOUR-STROKE CYCLE

drawn into the cylinder. This event is shown by the line *ab* of the indicator diagram, Fig. 211b. On its return stroke to the left, both intake and exhaust valves remaining closed, the mixture is compressed, nearly adiabatically, along the line *bc*. About 10 or 15 degrees of crank angle before *c* is reached, the electric spark to kindle the charge is produced. The timing of the spark and the explosive action are so related that the pressure rise produces the straight vertical line *cd* on the indicator diagram,

i.e. the explosion and resulting pressure rise occur at *constant volume*. During the next forward stroke of the piston, the force produced by the expanding gases of combustion develops the useful power of the engine as indicated by line *de*. When the power stroke is about 95 per cent completed, a cam driven from the crankshaft forces the exhaust valve open and release occurs. During the next piston stroke, the burned gases are swept from the cylinder through the exhaust valve as indicated by the line *ea* at a pressure slightly above atmospheric. This completes the cycle, which is repeated as long as the engine is operating.

The pressure scale of the intake and exhaust lines near the bottom of Fig. 214*b* is somewhat exaggerated in order to show the negative-work area *afb* which is produced by the pumping action of the piston on the intake and exhaust gases. This loss is small but appreciable.

It is thus seen that there is only one power stroke for every two revolutions of the crankshaft. The number of power impulses may be increased by connecting more cylinders and pistons to the crankshaft. As many as 6, 8, 12, or even 24 cylinders are employed in marine, automotive, and aeronautical engines.

215. The Two-Stroke Cycle. In the two-cycle engine which was first successfully developed by Dugald Clerk, the four events of the four-cycle engine occur within the time required to complete one revolution or two piston strokes. These four events which must occur during each cycle are: (1) intake of a fresh charge of the air-fuel mixture; (2) compression of the charge; (3) ignition and expansion of the burned charge; and (4) expulsion of the products of combustion.

The two-cycle engine is shown in diagrammatic form in Fig. 215. It consists of cylinder, piston, crank, and connecting rod as is the case in the four-cycle type. But it differs in two respects. First, the crank is enclosed in an air-tight case into which the fresh charge is drawn through the spring-loaded valve *V*. The crankcase and lower side of the piston function as a pump which draws in and compresses the properly mixed charge of fuel and air. The crankcase pressure usually ranges from one to five pounds per

square inch above atmospheric. Second, the piston P serves as a valve to open and close in the proper sequence the intake port C and the exhaust port E . In some designs the intake ports are located slightly nearer the crank end of the cylinder than the exhaust port (see Figs. 257 and 261). On the side of the piston

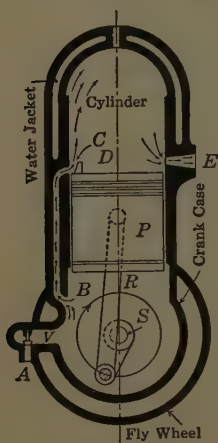


FIG. 215. — TWO-CYCLE ENGINE

head adjacent to the intake port, a guide is sometimes placed which aids in deflecting the incoming charge to the end of the cylinder and forcing the burned gases more completely from the cylinder. There is always some diffusion of the fresh charge and the burned gases, and some of the fresh charge escapes through the exhaust port, all of which combine to reduce the power output and increase the fuel consumption of the engine. The loss of fuel into the exhaust during the scavenging period seems to be unavoidable if the scavenging of the burned gases is to be sufficiently thorough. This is a serious defect in gas and gasoline engines which is not present in Diesel two-

cycle engines, because only fresh air is blown into and through the Diesel cylinder during the scavenging period.

The operation of the engine is as follows: Beginning with the piston near the crank end of its stroke, but moving upward as indicated in Fig. 215, the slightly compressed mixture of fuel and air flows from the crankcase into the cylinder through the passage BC . This continues until the piston rises to close the port C . Meanwhile the burned gases continue to flow out the exhaust port E until it in turn is closed by the rising piston when compression begins. Ignition is started 10 or 15 degrees of crank angle before the top end of the stroke is reached, and combustion occurs at practically *constant volume* as in the case of the four-cycle engine. The exploded gases expand producing useful power during the down piston stroke. At the completion of 80 to 90 per cent of the down stroke, the piston uncovers the exhaust port, and expulsion of the burned gases begins and continues until

the piston closes it again on the next upward stroke. During the up stroke, a new charge of air-fuel mixture is drawn into the crankcase through the valve *V*. It is compressed to a pressure varying from one to five pounds per square inch on the succeeding down stroke. The pressure-volume diagram for the work end of the cylinder is shown in Fig. 216. Its area represents to scale the foot-pounds of work done on the piston during one cycle of two piston strokes or one revolution of the crankshaft. A somewhat similar diagram may be obtained by attaching an indicator to the crankcase. But the work done in the crankcase is only from 5 to

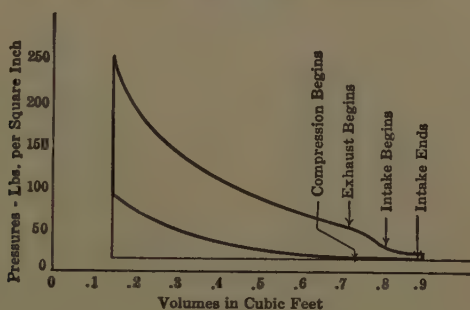


FIG. 216.—INDICATOR DIAGRAM FOR TWO-CYCLE ENGINE

10 per cent of the work done in the cylinder. However, the crankcase I. hp. must be deducted from the I. hp. of the cylinder to obtain the net I. hp. of the engine. Due to the crankcase work (or in large engines due to the power absorbed by separate scavenging pumps), and due to dilution of the fresh charge by the burned gases which are not completely expelled from the cylinder, the developed horsepower of a two-cycle engine is only from 1.6 to 1.8 times the developed power of a four-cycle engine having the same speed and dimensions.

In the early days, the two-cycle engine was employed almost exclusively in the marine field, where the engine weight is of prime importance. But as the two-cycle has been improved greatly in its fuel rate and its reliability, this style is rapidly driving the four-cycle engine from stationary service, especially in the larger sizes.

216. Internal-Combustion-Engine Fuels. During the early developmental period of the internal-combustion engine, manufactured coal gas was the only fuel used in them. No large engines were built; hence the aggregate fuel consumption was small compared to modern times. Since the arrival of the automobile and the Diesel engine, the bulk of the fuel used is in the liquid state. It is obtained largely from petroleum and its products. Table 66 contains a list of the fuels in more or less general use in I.C. engines at the present time.

TABLE 66
INTERNAL-COMBUSTION-ENGINE FUELS ¹

GAS	{	Natural gas	
	{	Manufactured	{ Illuminating gas { Producer gas
	{	By-product	{ Coke-oven gas { Blast-furnace gas
LIQUID	{	Petroleum	{ Gasoline { Diesel-engine oil { Fuel oil
	{	Alcohol	
SOLID	{	Pulverized coal	
	{	Flour, etc.	

In the vicinity of oil fields and a few other favored localities, natural gas in sufficient quantities to be commercially profitable is available for power generation. Relatively little power is produced from manufactured gas at the present time. From 1900 to 1910 a considerable amount of power was produced by blast-furnace gas in large gas engines (500 to 2000 horsepower units), but much difficulty was experienced in cleaning the gas. Much of the by-product gas is now burned under steam boilers. By far the greater portion of all internal-combustion-engine power is now generated from petroleum and its products.

Alcohol is used as a fuel only to a slight extent but it has great

¹ Typical analyses of the fuels will be found in Table 20, p. 168; Table 27, p. 191; and Table 28, p. 195.

potentialities for the future when the supply of petroleum becomes exhausted, as it surely will. As yet the black race which inhabits principally the torrid zone has contributed but little to advance and to carry on the work of civilization. It may well happen that within a few decades this race will be largely employed in growing suitable coarse vegetation on a large scale in favored localities of the torrid zone for the production of alcohol, which will then be used in internal-combustion engines to generate the world's power.

Solid fuels in pulverized form may also become an important internal-combustion-engine fuel as the store of petroleum in the earth's surface becomes exhausted. It was the opinion of certain coal-mining men extending through the 19th century that coal dust was explosive under favorable conditions. This fact was definitely proved at the U. S. Geological Survey testing station at Pittsburgh, Pa., during 1908-1909, and since that time it has been accepted by all scientists and engineers as an established phenomenon.¹ Attempts, not commercially successful as yet, have been made to use pulverized coal, flour, etc., for fuel in I.C. engines. There is reason to believe that such fuels may eventually be successfully used in either the Otto engine or the Diesel engine or perhaps both. It may be recalled here that Diesel attempted to operate his first engine on pulverized coal. Experimental work has been started on an ordinary automobile engine by the U. S. Bureau of Chemistry employing floor sweepings of grain elevators. Two Diesel engines, one a single-cylinder developing 80 horsepower and the other a three-cylinder engine developing 180 horsepower have been in successful operation for a number of years at Gorlitz, Germany, burning pulverized coal.²

217. Air-Standard Efficiency for the Ideal Otto Cycle. The air-standard efficiency has had a very wide acceptance in the engineering world. It was first officially proposed in 1905 by a Committee of the Institution of Civil Engineers of Great Britain

¹ See *U.S.G.S. Bulletin* 333, 1907; and *Bureau of Mines Bulletins* 20, 1911; 50, 1913; and 102, 1916.

² W. A. Noel and R. Hellback, *An Engine that Runs on Dust*, Power, Sept. 14, 1926, p. 402; Nov. 6, 1928, pp. 746 and 748.

which was appointed to study the Standards of Efficiency of Internal Combustion Engines.¹

The efficiency proposed by this committee was based upon the following five assumptions which greatly simplify the problem: (1) that the combustion occurs at constant volume, and is complete at the instant of maximum pressure; (2) that there is no transfer of heat during the compression and expansion periods, *i.e.* these changes are strictly adiabatic; (3) that the medium of heat exchange is atmospheric air considered to be a perfect gas, and its chemical composition remains unchanged throughout the engine cycle; (4) that the values of specific heats c_p and c_v , and their ratio k , are constant at all temperatures and pressures throughout the cycle; (5) that at a given temperature and pressure the specific volume of the medium is the same before and after the heat addition. Hence in the ideal Otto or constant-volume cycle, both the addition and rejection of heat occur at constant volume along the lines ab and cd of Fig. 217. Therefore

$$V_a = V_b$$

and

$$V_d = V_c.$$

The ratio of expansion (also known as the compression ratio), always a number greater than unity, equals

$$(508) \quad r = \frac{V_d}{V_a} = \frac{V_c}{V_b} = \left(\frac{T_a}{T_d} \right)^{1/(k-1)} = \left(\frac{T_b}{T_c} \right)^{1/(k-1)}.$$

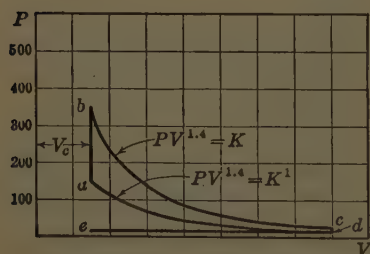


FIG. 217.—IDEAL OTTO CYCLE

Then

$$(509) \quad T_a = T_d r^{k-1},$$

and

$$(510) \quad T_b = T_c r^{k-1}.$$

Then the heat added for each pound of the substance equals

$$(511) \quad H_1 = c_v(T_b - T_a),$$

The heat rejected per pound of the substance is

$$(512) \quad H_2 = c_v(T_c - T_d).$$

¹ See Proc. Inst. C. E., vol. 162, 1905, pp. 307-338.

The efficiency of the Otto cycle, as in the case of the Carnot cycle, equals

$$(513) \quad e_0 = \frac{\text{heat received} - \text{heat rejected}}{\text{heat received}} = \frac{H_1 - H_2}{H_1}.$$

Substituting (511) and (512) in (513) one finds that

$$(514) \quad e_0 = \frac{c_v(T_b - T_a) - c_v(T_c - T_d)}{c_v(T_b - T_a)} = 1 - \frac{T_c - T_d}{T_b - T_a}$$

(508) may be written

$$(515) \quad \frac{T_d}{T_a} = \frac{T_c}{T_b} = \frac{1}{r^{k-1}}.$$

By division (515) becomes

$$(516) \quad \frac{T_c - T_d}{T_b - T_a} = \frac{T_d}{T_a} = \frac{T_c}{T_b} = \frac{1}{r^{k-1}}.$$

Hence substituting (516) in (514), one obtains

$$(517) \quad e_0 = 1 - \frac{T_d}{T_a} = 1 - \frac{T_c}{T_b} = 1 - \left(\frac{1}{r}\right)^{k-1}.$$

The first two expressions for efficiency in (517) may be restated thus

$$(518) \quad e_0 = \frac{T_a - T_d}{T_a} = \frac{T_b - T_c}{T_b}.$$

It will be noted that in these forms the expression for efficiency is identical to that for the Carnot engine. It will be further noted that the efficiency of the ideal constant-volume engine is independent of the nature of the medium of the heat exchanges as is the case for the Carnot cycle. Many attempts have been made to measure the various temperatures occurring in the cycle of actual Otto engines. But with the cycle being repeated from 100 to 3000 times per minute, no satisfactory means for measuring these temperatures accurately have yet been developed. Hence the expression from (519) as given below is used almost entirely in computing the ideal efficiency of the Otto engine.

$$(519) \quad e_0 = 1 - \left(\frac{1}{r}\right)^{k-1}.$$

The value of the efficiency computed by (519) depends only upon the ratio of compression and the value of k . The value of r required in evaluating (519) is readily obtained from the cylinder dimensions, and the value of k for air at atmospheric conditions is 1.403. But for all practical purposes the number 1.4 is sufficiently accurate. Then (519) becomes

$$(520) \quad e_0 = 1 - \left(\frac{1}{r}\right)^k.$$

The results obtained by solving (520) are known as *air-standard efficiencies*.

Considerable progress in determining the values of c_p , c_v , and k for temperatures ranging up to approximately 3000° F. has been made (see § 56, p. 74) at or near atmospheric pressure. But as yet very little information is available showing the changes in the values at high pressures. Much effort and study has been expended in recent years in attempting to determine the efficiency of the actual engine cycle, taking into account the variable specific heats and the varying composition of the mixture in the cylinder during the various stages of the cycle.¹ By applying the best information at present available regarding variable values of the specific heats, and by computing constants for the actual gaseous mixtures which are formed during the successive steps of the cycle, efficiencies designated as real-mixture standard are obtained. These new efficiencies, for corresponding compression ratios, vary in value from 70 to 85 per cent of the air-standard values obtained by equation (520). Since there is no universally accepted formula for computing efficiencies by the real-mixture standard, any attempt to develop the method is not undertaken here.

218. Air-Standard Efficiency for the Ideal Diesel Cycle. In the following development the subscripts indicate values of the various properties at the points indicated in Fig. 218. The

¹ See G. A. Goodenough, *Principles of Thermodynamics*, 3d ed., pp. 268–313; and *Bulletin No. 139*, University of Illinois, 1924.

Telford Petrie, *Elements of Internal Combustion Engineering*, pp. 97–100.

R. C. H. Heck, *Ideal Gas-Engine Cycles*, Transactions A.S.M.E., vol. 48 1926, pp. 1029–1073.

F. O. Ellenwood, F. C. Evans, and C. T. Chwang, *Efficiencies of Otto and Diesel Engines*, Transactions A.S.M.E., Jan.–Apr., 1928.

events of the ideal four-stroke Diesel cycle are represented by the following lines:

ed, intake of charge of cold air only at atmospheric pressure,
da, adiabatic compression of the air to a pressure of from 400 to 700 lbs. per sq. in. and the corresponding temperatures varying from about 1000° F. to 1400° F.,

ab, fuel injection and combustion at such rates as to maintain constant pressure from *a* to *b*, the temperature rising to approximately 3000° F. at *b*,

bc, adiabatic expansion; the value of *k* is assumed to be the same for both *da* and *bc*,

cd, release of the burned gases and pressure falling at constant volume to atmospheric,

de, expelling the remainder of the burned gases from the cylinder at atmospheric pressure.

In this cycle heat is added at *constant pressure* along *ab*, and heat is rejected at *constant volume* along *cd*. It is also assumed that the medium of heat exchange is a perfect gas having the same physical constants as air at atmospheric conditions; as was done in the case of the Otto-cycle engine. This means that c_p , c_v , and c_p/c_v have constant values at all points throughout the cycle. Then the thermal efficiency equals

$$(521) \quad e_d = \frac{\text{heat received} - \text{heat rejected}}{\text{heat received}} = \frac{H_1 - H_2}{H_1}.$$

Then for one pound of the working substance

$$(522) \quad \begin{aligned} e_d &= \frac{c_p(T_b - T_a) - c_v(T_c - T_d)}{c_p(T_b - T_a)} \\ &= 1 - \frac{c_v}{c_p} \left(\frac{T_c - T_d}{T_b - T_a} \right). \end{aligned}$$

Let

$$r_1 = \text{compression ratio} = \frac{V_d}{V_a},$$

and

$$r_2 = \text{expansion ratio} = \frac{V_c}{V_b}.$$

During adiabatic compression,

$$(523) \quad T_a = T_d \left(\frac{V_d}{V_a} \right)^{k-1} = T_d r_1^{k-1}.$$

During fuel injection along ab , the equation of state holds with the result that

$$(524) \quad \frac{P_a V_a}{T_a} = \frac{P_b V_b}{T_b}.$$

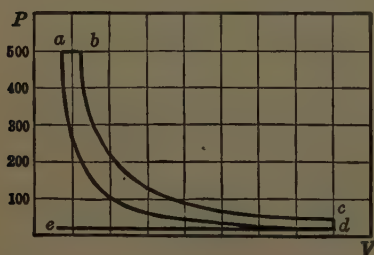


FIG. 218.—IDEAL DIESEL CYCLE

But

$$P_a = P_b,$$

hence

$$(525) \quad \frac{V_a}{T_a} = \frac{V_b}{T_b},$$

and

$$(526) \quad \frac{T_b}{T_a} = \frac{V_b}{V_a}.$$

The second member of (526) may be multiplied by V_d/V_d , or unity, without changing its value, hence

$$(527) \quad \frac{V_b}{V_a} = \frac{V_b}{V_a} \times \frac{V_d}{V_d} = \frac{V_d}{V_a} \times \frac{V_b}{V_d} = \frac{r_1}{r_2}.$$

Therefore (526) becomes

$$(528) \quad T_b = T_a \frac{r_1}{r_2}.$$

Substituting the value of T_a from (523) in (528), one has

$$(529) \quad T_b = \frac{T_d r_1}{r_2}.$$

Along the expansion curve the following relation holds:

$$(530) \quad T_c = T_b \left(\frac{V_c}{V_b} \right)^{k-1} = \frac{T_b}{r_2^{k-1}}.$$

Substituting (529) in (530) one finds

$$(531) \quad T_c = T_d \left(\frac{r_1}{r_2} \right)^k.$$

From (523), (529), and (531) the following is obtained

$$(532) \quad \frac{T_c - T_d}{T_b - T_a} = \frac{\left(\frac{r_1}{r_2}\right)^k - 1}{\frac{r_1^k}{r_2} - r_1^{k-1}} = \frac{\left(\frac{r_1}{r_2}\right)^k - 1}{r_1^{k-1} \left[\frac{r_1}{r_2} - 1 \right]}.$$

Remembering that $c_v/c_p = 1/k$ and substituting (532) in (522), one finds that

$$(533) \quad e_d = 1 - \frac{1}{k} \left[\frac{\left(\frac{r_1}{r_2}\right)^k - 1}{r_1^{k-1} \left(\frac{r_1}{r_2} - 1\right)} \right].$$

Substituting the value of k for air equal to 1.4 and removing the factor r^{k-1} from the brackets, one obtains

$$(534) \quad e_d = 1 - \left[\frac{1}{r} \right]^{0.4} \left[\frac{z^{1.4} - 1}{1.4(z - 1)} \right],$$

in which

$$z = \frac{r_1}{r_2} = \frac{V_d}{V_a} \div \frac{V_d}{V_b} = \frac{V_b}{V_a}.$$

An examination of (534) shows that it consists of two parts. The first is the expression for the efficiency of the ideal Otto engine. The second is a number whose value is always greater than unity. Hence for equal compression ratios, the air-standard efficiency of the Diesel engine is smaller than it is for the Otto engine (see Fig. 219). Practically, however, the Otto engine efficiency is necessarily always lower because the permis-

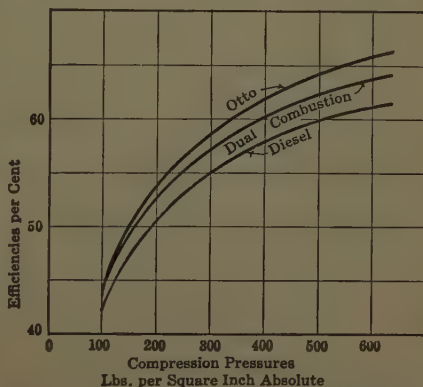


FIG. 219.—AIR-STANDARD EFFICIENCIES

sible compression ratio is limited by the minimum compression pressure which will produce spontaneous ignition of the air-fuel mixture; while the efficiency of the Diesel engine is not so limited. The limits of compression pressure in the Otto engine, fixed by the properties of the fuel, vary from about 40 lbs. per sq. in. gage for kerosene, and 75 lbs. for gasoline, up to about 200 lbs. for ethyl alcohol. In the Diesel engine, there are no limits fixed by the fuel, but considerations of strength, safety, and reasonably long life of the engine fix the upper pressure at the end of compression at 600 or 700 lbs. per sq. in.

219. The Dual-Combustion Cycle and Air-Standard Efficiency.

This cycle in the ideal case may be considered as following partly the Otto and partly the Diesel ideal cycles as indicated in Fig. 220. The distinctive feature of the dual-combustion cycle is that heat is generated partly at constant volume and partly at constant pressure. All of the so-called semi-Diesel or pre-combustion and solid-injection Diesel engines operate on modifications of this cycle. The efficiency of the ideal cycle was first worked out by W. J. Walker.¹

Employing the notation of sections 217 and 218, the ideal efficiency e_s (the subscript s indicates semi-Diesel or Dual combustion cycle) is found as follows. Similar to equation (521), the efficiency for unit weight of gas equals

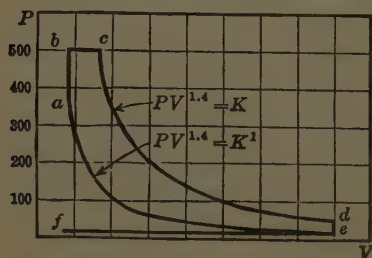


FIG. 220.—DUAL-COMBUSTION CYCLE

$$(535) \quad e_s = \frac{H_1 - H_2}{H_1}.$$

Since heat is added at both constant volume and constant pressure

$$(536) \quad H_1 = c_v(T_b - T_a) + c_p(T_c - T_b).$$

As in the case of the ideal Diesel engine, heat is rejected at constant volume

$$(537) \quad H_2 = c_v(T_d - T_e),$$

¹ See Philosophical Magazine, vol. 34, 1917, p. 168; and Proceedings of the Institution of Mechanical Engineers, Dec., 1920, p. 1235.

and

$$(538) \quad e_s = 1 - \frac{c_v(T_d - T_e)}{c_v(T_b - T_a) + c_p(T_c - T_b)}.$$

Dividing both members of the fraction of (538) by c_v one obtains

$$(539) \quad e_s = 1 - \frac{T_d - T_e}{(T_b - T_a) + k(T_c - T_b)}.$$

Let

$$r_1 = \frac{V_e}{V_a} = \frac{V_d}{V_a} = \text{ratio of compression,}$$

$$r_2 = \frac{V_d}{V_c} = \frac{V_e}{V_c} = \text{ratio of expansion,}$$

$$r_3 = \frac{P_b}{P_a} = \text{ratio of combustion pressures,}$$

$$r_4 = \frac{T_c}{T_b} = \frac{V_c}{V_b} = \text{ratio combustion volumes.}$$

All the temperatures in (538) may be expressed in terms of T_e and the above four ratios as follows:

$$(540) \quad T_a = T_e r_1^{k-1},$$

$$(541) \quad T_b = T_a r_3 = T_e r_3 r_1^{k-1},$$

$$(542) \quad T_c = T_b r_4 = T_e r_3 r_4 r_1^{k-1},$$

$$(543) \quad \begin{aligned} T_d &= T_c \left(\frac{1}{r_2} \right)^{k-1} = T_e \left(\frac{1}{r_1} \right)^{k-1} r_4^{k-1}, \\ &= T_e \left(\frac{1}{r_1} \right)^{k-1} r_4 r_4^{k-1} r_3 r_1^{k-1} = T_e r_3 r_4^k. \end{aligned}$$

Substituting the values from (540) to (543) inclusive in (539), one finds that

$$(544) \quad \begin{aligned} e_s &= 1 - \frac{r_3 r_4^k - 1}{(r_3 r_1^{k-1} - r_1^{k-1}) + k(r_3 r_4 r_1^{k-1} - r_3 r_1^{k-1})} \\ &= 1 - \left[\frac{1}{r_1} \right]^{k-1} \left[\frac{r_3 r_4^k - 1}{(r_3 - 1) + k r_3 (r_4 - 1)} \right]. \end{aligned}$$

An analysis of equation (544) shows that for the special case when $r_3 = 1$ the factor in the brackets equals the bracket factor

for the Diesel cycle (equation (534)). And when the value of $r_4 = 1$ the value of the bracket reduces to unity which is the case for the Otto cycle. Hence it is seen that (544) may be considered as the general equation for the air-standard efficiency for all classes of internal-combustion engines.

Comparison of the values of e_0 , e_d , and e_s is shown by the curves of Fig. 219 in which the values for e_s have been calculated on the following assumptions:

- (1) $p_b = p_a + 150$ lbs. per sq. in.,
- (2) $V_c = V_a + 3$ per cent of piston displacement.

It is seen that the values of efficiency for the dual-combustion cycle approach closely those for the Otto cycle.

220. Power and Efficiencies of Internal-Combustion Engines.

The indicated horsepower of an actual I.C. engine is obtained by attaching an indicator to the cylinder in a manner similar to that for steam engines and measuring the power from the indicator cards. The useful power or the developed horsepower must be measured by means of some kind of a brake or other loading arrangement whose power can be measured, as an electrical generator whose performance curves have been determined. (See Fig. 256 for actual indicator cards.) Let

a = net area of the piston in sq. in.,

C = number of active combustion chambers in the engine,

l = length of piston stroke in feet,

n = r.p.m. of engine shaft,

p = mean effective pressure from the indicator-diagram,
lbs. per sq. in.,

s = piston speed, feet per min.,

I.hp. = indicated horsepower of the engine,

D.hp. = developed horsepower of the engine,

e_m = mechanical efficiency,

e_0 = Otto cycle air-standard efficiency,

e_d = Diesel cycle air-standard efficiency,

e_t = thermal efficiency based on D.hp.,

e_p = process efficiency.

Then for all four-stroke-cycle single-acting engines,

$$(545) \quad \text{I.hp.} = \frac{\text{Casp}}{4 \times 33,000} = \frac{\text{Casp}}{132,000}.$$

For all two-stroke-cycle, single-acting engines

$$(546) \quad \text{I.hp.} = \frac{\text{Casp}}{2 \times 33,000} = \frac{\text{Casp}}{66,000}.$$

In (535) and (536)

$$(547) \quad s = 2 \ln.$$

$$(548) \quad e_m = \frac{\text{D.hp.}}{\text{I.hp.}},$$

$$(549) \quad e_t = \frac{2543}{\text{B.t.u. supplies per D.hp. per hr.}},$$

and

$$(550) \quad e_p = \frac{e_t}{e_0} \quad \text{or} \quad \frac{e_t}{e_d},$$

as the case may require.

221. Ignition or Kindling Temperatures of Fuels. Oxidation or combustion of many fuels is taking place at very slow rates even at ordinary atmospheric conditions. As the temperature of the fuel and air is raised, the velocity of the reaction increases. Many empirical laws indicating the relation between the initial temperature and the rate of combustion have been proposed by different investigators. Van't Hoff stated that for each 10°C . increase, the combustion rate doubles. A law widely accepted is that the velocity of combustion is nearly proportional to the square root of the absolute temperature.

Consider the case of a combustible mixture of a gaseous fuel and air made up of a small central sphere surrounded by a series of thin spherical shells. Suppose the sphere is heated to some temperature $a^\circ \text{F}$.; thus combustion is induced at a certain rate. The resultant heat will be conducted and radiated to the adjacent spherical lamina. But the rate of heat generation is not sufficient to raise the temperature of this adjacent lamina at $a^\circ \text{F}$. The combustion in that spherical shell will not be as rapid as in the

central sphere. Likewise, throughout the successive outer shells the combustion rate will continue to decrease until it falls to the rate due to the temperature of the outermost shell which may be supposed to be at atmospheric temperature. Again, consider the central sphere of mixture to be raised to a temperature C° F. at which the rate of combustion is very high, and heat is generated so fast that the temperature of the first adjacent shell is raised to a temperature higher than C° F. The rate of combustion in this shell will be greater than in the central sphere. The combustion rate will then accelerate from shell to shell until a maximum temperature is reached. There must be some temperature b° F., between a° F. and C° F., at which the combustion rate will just be sufficient to raise the temperature of each concentric shell to the temperature b° F. This temperature b° F. is the *ignition temperature* or *kindling temperature* of the mixture. As soon as any portion of the mixture attains the temperature b° F., the combustion becomes self-sustaining, and inflammation spreads throughout the mixture.

From the above it may be inferred that the ignition temperature will vary with different fuels, with various proportions of the fuels in the mixture, and with the pressure to which the mixture is subject, and in general the higher the pressure the lower will be the ignition temperature. Because at higher pressures the molecules of the mixture will be forced closer together, and the opportunity to lose heat by radiation will be less. It can be further inferred that only certain limiting proportions of combustible gases and air will support a sufficiently rapid combustion to be self-sustaining, because the molecules of the gas which are in excess will carry away so much heat energy that no possible initial heating will be able to produce the transmission of a kindling temperature through the mixture. The limiting proportions within which inflammation of several fuels can be induced at ordinary room temperatures are given in Table 67.¹ The limits of inflammation generally widen out as the initial temperature of the mixture rises.

¹ For a more complete table see Marks' *Mechanical Engineers' Handbook*, 2d edition, p. 1096; and Bureau of Mines Technical Paper No. 427.

TABLE 67

LIMITS OF INFLAMMABILITY OF AIR-GAS MIXTURES
(Gas in the Mixture, per Cents by Volume)

FUEL	LOW GAS LIMIT	HIGH GAS LIMIT	FUEL	LOW GAS LIMIT	HIGH GAS LIMIT
Hydrogen.....	9.0	67.0	Gasoline. { 73° B 60° B	1.9	5.2
Oil gas.....	7.0	18.0		1.5	
Methane.....	6.0	13.0			

Curves for several hydrocarbons showing the relation between the air pressure and the ignition temperature are shown in Fig. 221. The results from which the curves are plotted were obtained in an experimental apparatus in which the conditions closely approximated those obtaining in Diesel engines.¹ It is seen that there is considerable reduction in the kindling temperature as the air pressure rises.²

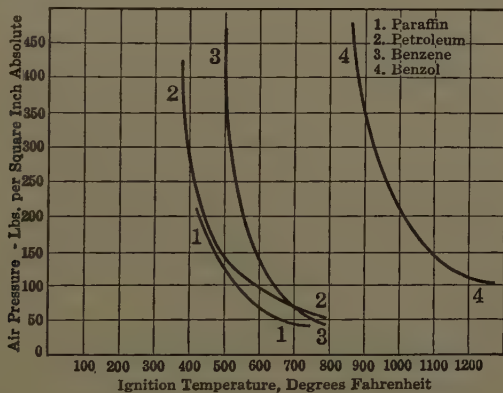


FIG. 221.—IGNITION TEMPERATURES

222. Detonation.³ By carrying the compression pressure sufficiently high, all fuels may be ignited automatically by the corresponding compression temperature. For maximum thermal efficiency in the Otto engine, the compression pressure should be the highest attainable. But it is always limited to the pressure

¹ See J. Tauss and F. Schutte, *Bestimmung des Zündpunktes unter Druck*, Zeitschrift des Vereines deutscher Ingenieure, May 31, 1924, pp. 574-578.

² Tables of kindling temperatures at atmospheric pressure for various fuels may be found in the engineering handbooks.

³ See E. Favary, *Motor Vehicle Engineering*, p. 335.

which produces the kindling temperature of the particular fuel under consideration. However, certain fuels, particularly light volatile hydrocarbons such as gasoline and kerosene, have a more or less pronounced tendency to ignite in an engine cylinder before the true kindling temperature is reached by compression alone. This phenomenon is known as detonation. Detonation preceeds and eventually produces automatic ignition with the lighter petroleum distillates. Those of the paraffin (C_nH_{2n+2}) series are most subject to detonation, while the aromatics (derivatives of C_6H_6) are least subject to it.¹

The tendency of a fuel to detonate may be reduced in a number of ways: (a) by weakening the mixture, (b) by introducing relatively large percentages of a non-detonating fuel such as alcohol or toluene (C_7H_8), (c) by incorporating very small percentages of substances which have been found by experience to greatly reduce detonation. Tetra-ethyl lead ($PbEt_4$) is the substance used in the ethyl gasoline of commerce for reducing the tendency to detonate, and permit the use of higher compression ratios with consequent higher thermal efficiencies.

Several theories have been proposed to explain the phenomenon of detonation. But none of them satisfactorily explain all the observed facts.² According to Ricardo's theory, which explains most of the observations relating to detonation, when combustion first starts it produces a rapid rise in the temperature and pressure in the cylinder. The rise takes place before the fuel is entirely burned. As soon as the temperature rises above the kindling point of the fuel, the entire remaining mixture is consumed almost instantly, producing a violent explosion. This sets up a pressure wave sufficiently rapid and powerful to produce the characteristic knock. This theory assumes that the fuel has a tendency to preignite under proper conditions. However, many fuels which detonate badly show no inclination to preignite, and vice versa.

¹ See H. R. Ricardo, *The Internal Combustion Engine*, vol. 2, 1923, p. 39.

² See G. L. Clark and W. C. Thee, *Theories of Detonation in Industrial and Engineering Chemistry*, vol. 17, Dec., 1925, pp. 1219-1226.

Proceedings of the Royal Society, A, vol. 114 (1927), pp. 137 and 152.

Proceedings of the Royal Society, A, vol. 116 (1927), pp. 516-529.

Proceedings of the Institute of Automobile Engineers, vol. 22 (1927-1928), pp. 233-270.

For instance, carbon bisulfide preignites readily but does not detonate. Kerosene is one of the worst offenders in this respect, but it does not readily preignite.

223. Ignition in Internal-Combustion Engines. An internal-combustion engine is a poor prime mover unless the charge can be surely and promptly ignited each time there is demand for a power impulse. Smooth operation of the engine, especially of the high-speed multicylinder type, demands an ignition system which insures a regular and uniform explosion. To be satisfactory, the system must be simple, efficient, and reliable under all conditions of operation. To develop such a system has been one of the difficult problems concerning combustion engines. At one time or another five ignition systems have been evolved. They are: (a) open flame, (b) hot tube, (c) catalytic and chemical action, (d) electric spark, and (e) spontaneous heat of compression.

(a) *Open-Flame Ignition.* Open-flame ignition seems to have been first used by Barnett and has already been briefly described. Various devices for flame ignition were in use as late as 1890. It was abandoned because of its unreliability and its inadequacy for high-speed multicylinder engines. Dugald Clerk's observations of 1886 on this topic are interesting. "It is somewhat humiliating to the inventor to watch a powerful gas engine at work, developing say 30 horses, and to know that he can at once change the whole and make the engine powerless by blowing out the external flame."¹ At that time, Clerk also predicted that the electric-ignition system would not survive because it was too complicated and unreliable.

(b) *Hot-Tube Ignition.* Ignition by means of a hot tube, heated to incandescence by an external flame, seems to have been proposed by Siemens. It was first reduced to practice about 1855 by A. V. Newton in England and Drake in America. It did not come into general use, however, until after 1880. This device continued to be used on small stationary engines as late as 1915.

The tube was generally made up of a piece of half-inch wrought-iron pipe, 6 to 12 inches long, closed off at the outer end by being

¹ Dugald Clerk, *The Gas Engine*.

forged to a point as indicated in Fig. 222. The closed end is heated above the kindling temperature of the fuel by an external gas flame or gasoline torch. The hot tube is protected from air

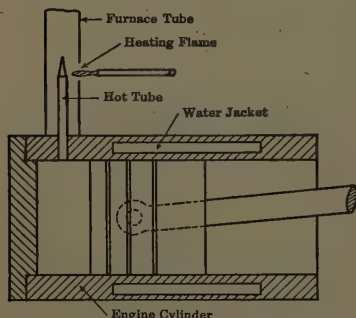


FIG. 222.—HOT-TUBE IGNITION

currents and excessive radiation by a tubular furnace surrounding it.

The time of the ignition in the cycle may be varied by changing the length of the tube or by moving the heating flame. As the tube-length increases, or the flame is moved farther out on the tube, the fresh charge has to travel farther as it is compressed into the tube to

reach the incandescent zone; hence the time of the explosion in the cylinder is retarded. However, the hot-tube ignition system does not permit of rapid or refined adjustment of the time of explosion since it depends upon several factors as follows:

- (1) Length of tube to incandescent zone;
- (2) Volume and length of passage leading to the tube;
- (3) Value of the end compression pressure;
- (4) The tube temperature; the hotter, the earlier the explosion;
- (5) Rotative speed of the engine;
- (6) Quality of the mixture in the cylinder.

(c) *Catalytic or Chemical Means of Ignition.* As one means of igniting the charge, Barnett employed sponge platinum contained in a small cup screwed into the walls of the cylinder head. The ignition is produced by catalytic action as hydrogen or illuminating gas is forced by compression into contact with the platinum. However, the timing of the explosion is uncertain, the platinum must be reactivated frequently, and at best its action is too slow for high speeds. Another chemical method used to a slight extent was to introduce a small quantity of phosphoretted hydrogen into the cylinder at the proper instant.

Its combustion automatically kindled the charge. None of the systems described above have survived. All have been replaced by electric-ignition systems which have been developed to a high degree of perfection.

(d) *Electric Ignition.* Two systems of electric ignition are in general use. They are:

- (1) Low-voltage make-and-break system,
- (2) High-voltage spark-gap system.

Electrical energy may be supplied from a number of sources as follows:

- (1) Primary batteries (wet or dry);
- (2) Secondary or storage batteries;
- (3) Direct-current generators (usually low-voltage) in connection with storage batteries;
- (4) Alternating-current generators (magnetos), which may be of the low-voltage or of the high-voltage type.

(e) *Low-Voltage Ignition.* The elements comprising the low-voltage, make-and-break system, and illustrated diagrammatically in Fig. 223, are a low-voltage generator (may be either alternating current or direct-current), or a primary battery, and inductance coil (also called reactance, kick, or retardation coil), and a device operated by an external mechanism to make and break the circuit within the compression space of the cylinder at the proper instants. One style of make-and-break device is

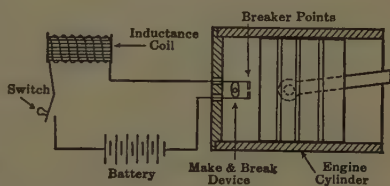


FIG. 223.—MAKE-AND-BREAK ELECTRIC IGNITION

shown in Fig. 224. It is inserted in the compression space through an opening in the cylinder head or the side walls of the cylinder. The entire unit may be removed for cleaning and adjusting by unscrewing the two bolts holding it in place.

The core of the inductance coil consists of a bundle of soft-iron wire possessing special magnetic properties so that the coil will magnetize and demagnetize rapidly without excessive energy loss.

A coil of insulated copper wire is placed on the core. When an electric current flowing through the coil is interrupted suddenly, a heavy spark of considerable intensity will be produced between the breaker points. The spark is due to the voltage induced by the rapidly changing magnetism in the coil when the



(Westinghouse Machine Co.)

FIG. 224.—MAKE-AND-BREAK DEVICE

circuit is broken. The battery is so connected in the circuit that its voltage is added to the induced voltage. The total voltage across the breaker points at the instant of separation usually varies from 100 to 500 volts, depending upon the design of the coil. The action of an inductance coil may, in a general way, be

likened to water hammer in a pipe line. When a valve at the down-stream end of a long line is closed suddenly, the inertia of the suddenly arrested stream will produce a forceful blow on the valve.

The make-and-break system is used only on slow-speed stationary and marine engines of one or two cylinders. The minimum time required to complete the make-and-break cycle is too great to be employed successfully on multicylinder high-speed automotive or airplane engines. However, in its proper field it is a simple reliable system which is much used.

(f) *High-Voltage Spark-Gap Electrical Ignition.* One modern system, illustrated diagrammatically in Fig. 225, frequently employed on automotive engines, consists of a low-voltage (6 or 12 volts) direct-current generator *A* connected in parallel through a low-voltage circuit breaker or relay to a storage battery *B*. The generator, driven from the engine, provides energy for ignition and lights when the engine is running. At automobile speeds above eight or ten miles per hour, the voltage generated is sufficient to close the relay, and the surplus energy is delivered to the storage battery. The battery is drawn upon for ignition energy when starting the engine, to operate a small starting

motor, and for lights when the engine is not running. The ignition system also includes a high-tension induction coil *D*, a circuit interrupter, and a spark plug.

The induction coil is a small high-ratio transformer, *i.e.* it has two entirely separate windings. The primary or low-voltage winding, connected in series with the battery and generator, contains a few hundred turns of insulated copper wire next the core. The wire ranges in size from No. 16 to No. 20 B & S gauge. The primary voltage is usually 6 or 12 volts. The secondary or

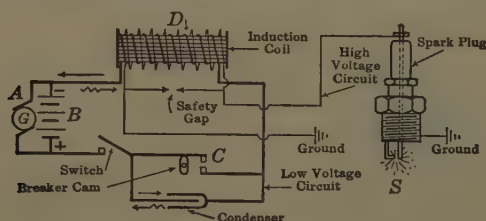


FIG. 225.—HIGH-VOLTAGE IGNITION SYSTEM

high-voltage winding, which is connected to the spark plug, consists of several thousand turns of enamel-insulated wire of about No. 36 B & S gauge. The two windings are carefully insulated from each other by paraffin or waxed paper. In order to thoroughly insulate the completed coil and render it moisture proof, it is usually imbedded completely in paraffin or some other insulating and moisture-proof compound.

An interrupter *c* is placed in the low-voltage circuit. A condenser is connected across the contact points of the interrupter. This greatly reduces sparking at the breaker points. The condenser discharge also aids in rapidly building up a high voltage in the secondary circuit. The interrupter points are normally held closed by a spring. They are opened by a cam driven from the engine cam shaft. The cam must be properly timed to open the contacts of the interrupter only when the engine piston is nearing the end of the compression stroke in order to ignite the charge at the proper instant. There must be as many lobes on the breaker cam as there are engine cylinders. Two views of an interrupter and distributor for a modern six-cylinder automobile engine are

shown in Fig. 226. The break is usually timed to produce the spark from 5 to 30 degrees of crank angle ahead of the dead-center position of the piston. The greater advance is used in the higher speed engines.

The spark plug is located within the compression space of the cylinder. It consists of two fixed terminals *S* with a fixed gap of about 0.025 inches between them. One terminal is heavily insulated by being imbedded in porcelain; the other is grounded to the cylinder. Sparking across the gap causes burning and

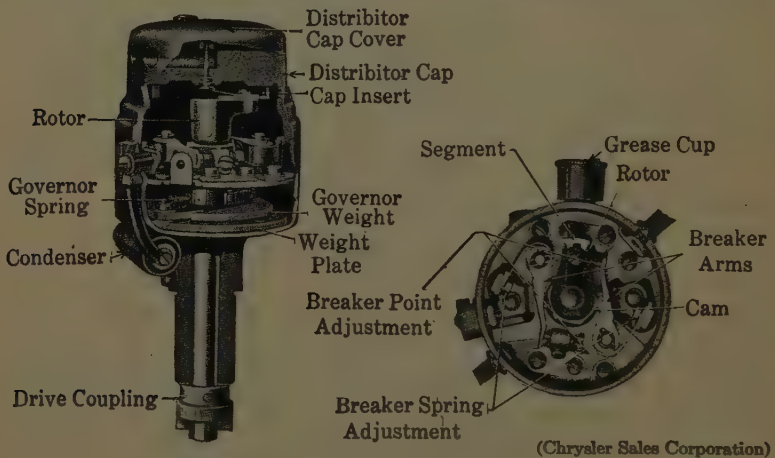


FIG. 226.—BREAKER AND DISTRIBUTER

pitting of the terminals. In earlier days they were made of platinum-iridium alloy, but its scarcity and high cost has made that impossible. Now the terminals are generally made of tungsten, which resists the pitting action nearly as well as the platinum alloy.

With the switch closed and the cam in such a position as to permit the interrupter points to make contact, current flows from the generator or battery through the primary winding, magnetizing the core. When the cam lobe separates the interrupter points, breaking the circuit, the core demagnetizes very rapidly. This sudden change in magnetization induces a current in both primary and secondary windings. By virtue of its many turns the voltage induced in the secondary windings is from 10,000 to 30,000 volts, causing a spark to jump across the gap *S*.

A voltage is induced in the secondary winding when the primary circuit is closed, as well as when it is broken. But the core magnetizes much more slowly than it demagnetizes, consequently the voltage generated, upon closure of the circuit, is not great enough to force a spark across the gap.

The condenser performs two functions. It prevents or greatly reduces undesirable sparking at the interrupter contacts, and it greatly increases the rate of demagnetization of the core. The latter aids in building up a greater voltage in the secondary winding. When the break in the primary circuit occurs, the induced surge of primary current which flows in the same direction as the battery current, and which would otherwise cause sparking at the interrupter points, rushes into the condenser, charging it. (This is indicated in Fig. 225 by arrows having straight shafts, thus $\rightarrow$.) The side of the condenser receiving the surge is momentarily charged positively. Almost instantly the condenser discharges back through the primary winding and battery in the opposite direction (This current is indicated by arrows with broken shafts, thus $\dashrightarrow$). Since the backward surge is in the opposite direction of the normal magnetizing current, it greatly hastens demagnetization thus building a much higher secondary voltage than would otherwise be possible. Actually, the secondary voltage oscillates and dies out rapidly due to the damping effect of the resistance and reactance of the circuit. The first surge is the only one having sufficient voltage to jump the spark-plug gap. A safety gap is usually provided across the high-tension winding. This gap is made slightly greater than the spark-plug gap, but it is small enough to permit a spark to pass the gap rather than to puncture the insulation, which would destroy it.

224. Modern Gasoline Engines. Modern gasoline engines which operate on the Otto cycle and employ a carburetor¹ to produce the correct air-gasoline mixture are employed as industrial, marine, automotive, and aeronautic prime movers. In all

¹ For description of carburetors and explanation of their functioning see A. B. Browne, *Handbook of Carburetion*; B. G. Elliott and E. L. Consoliver, *The Gasoline Automobile*; H. R. Ricardo, *The Internal Combustion Engine*.

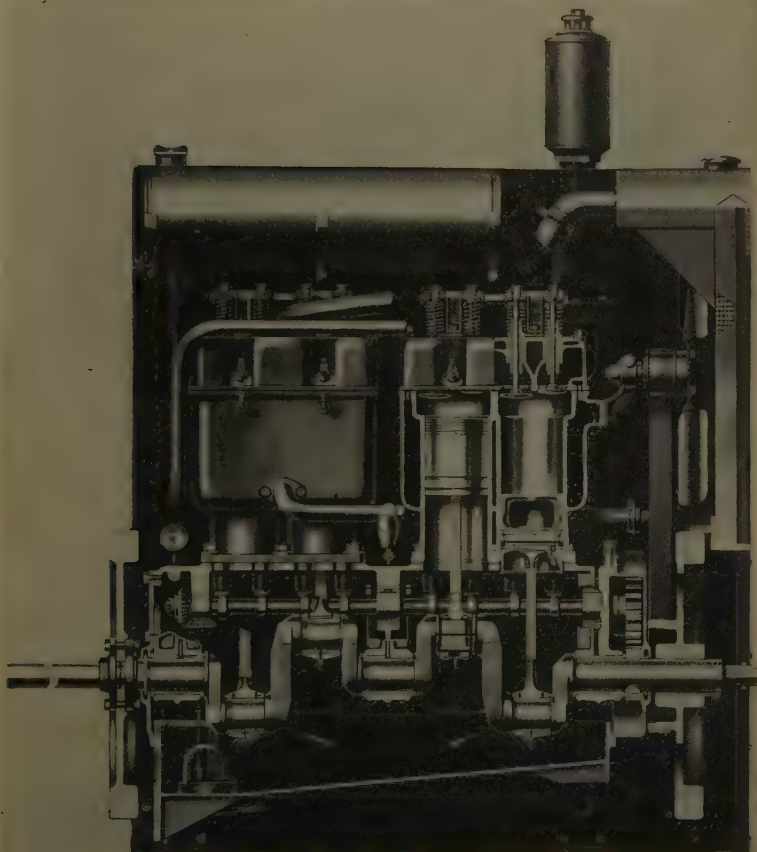
these fields of service, the vertical, single, and multicylinder, high-speed (400 to 7000 r.p.m.), four-stroke cycle designs prevail; although many two-stroke cycle engines are employed for marine work on account of their smaller weight. But in the automotive and aeronautical fields the four-cycle type is employed exclusively on account of its greater reliability and better fuel economy. They are made up with from one to twelve cylinders operating on a single crankshaft (one manufacturer is building 24-cylinder engines) and the power capacity ranges all the way from a fractional horsepower to a maximum of about 1000 horsepower.

225. The Novo Industrial Engines. A four-cylinder four-cycle heavy-duty automotive-type industrial gasoline engine built by the Novo Engine Company is shown in partial section by Fig. 227. This style of engine is built in two sizes rated at 40 and 50 brake horsepower at 925 r.p.m. The principal specifications of the engine are given in Table 68.

TABLE 68
SPECIFICATIONS FOR NOVO FOUR-CYLINDER ENGINES

	E. F. ENGINES	X. F. ENGINES
Rated brake horsepower.....	40	50
Rated r.p.m.....	925	925
Cylinder bore (inches).....	4.75	5.25
Stroke (inches).....	6.00	6.00
Main bearings.....	2.5 in. diam. by 4 in. long	
Connecting-rod bearings.....	2.5 in. diam. by 3 in. long	
Piston pins.....	1.375 in. diam., hardened and ground	
Piston rings.....	3 per piston, 0.35 in. wide	
Carburetor.....	Zenith, 1.5 in.	
Ignition system.....	Splitdorf high-tension magneto with impulse coupling	
Cooling system.....	Radiator, fan, and water circulating pump	
Fuel.....	Gasoline, gas, or kerosene	
Flywheel.....	Two per engine, diam. 22 in.	
Engine weight incl. flywheel (lbs.).....	1550	1650
Weight per B.hp. (lbs.).....	39	33
Specific gasoline rate lbs. per B.hp. per hr.....	0.64 lbs. at full load	

The engines are of the valve-in-head type. The cylinders are cast in pairs, and bolted to the crankcase. They are provided with removable water-cooled heads. The crankshaft is carried in three main bearings. The engine is regularly arranged for



(Novo Engine Co.)

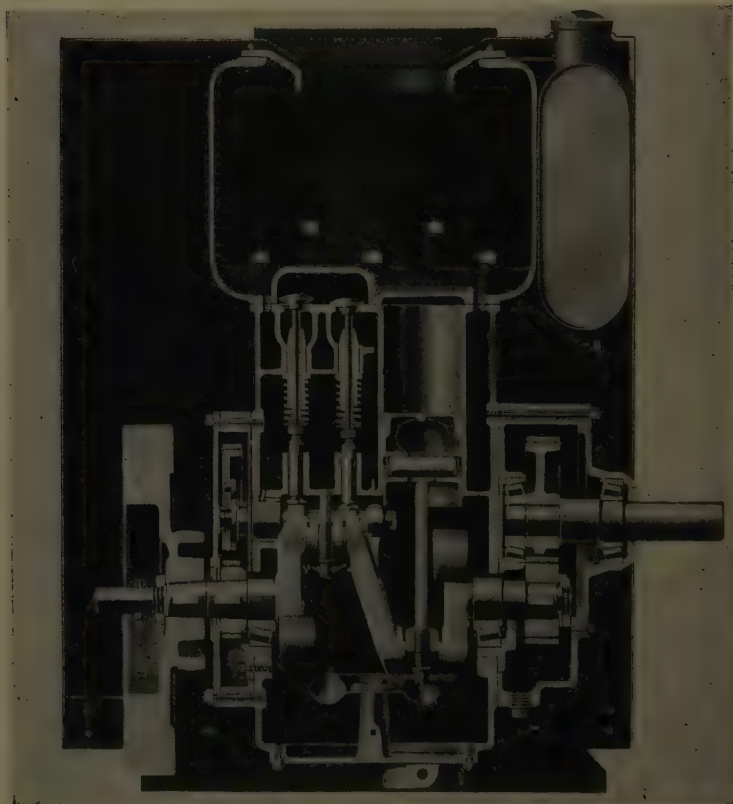
FIG. 227.—AUTOMOTIVE-TYPE INDUSTRIAL ENGINE

direct connection through a flexible coupling to the driven units, or it may be equipped with a clutch and geared speed reduction.

The engine speed is controlled by a centrifugal governor which actuates a butterfly throttle valve in the intake manifold. The

engine is usually started with a hand crank, but electric starting will be provided on special order.

The complete engine is enclosed in a steel housing which may be mounted on a truck, thus providing a light weight, portable unit for driving air compressors, concrete mixers, hoists, and pumps on construction work.



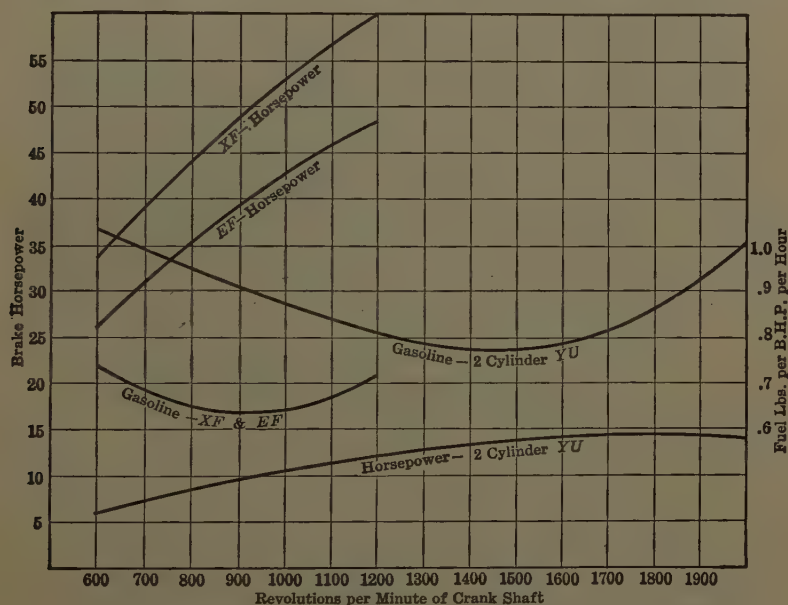
(Novo Engine Co.)

FIG. 228.—TWO-CYLINDER INDUSTRIAL ENGINE

A Novo two-cylinder gasoline engine built in sizes from 6 to 12 horsepower is shown in Fig. 228. In many respects this engine is very similar to the four-cylinder style of Fig. 227. It differs in that the crankshaft and driveshaft are equipped with Timken roller bearings. Cooling is accomplished by thermosiphon action

from a hopper of water attached to the cylinder heads, or it may be accomplished by radiator and circulating-water pump if desired. The engine may be arranged for direct drive with a clutch or through gear reduction with or without clutch. Single-cylinder engines and four-cylinder engines equipped with Timken roller bearings of this same general design are available also.

Performance curves for these engines are shown in Fig. 229. At 1200 r.p.m. the engines are capable of delivering a maximum of 48 and 60 brake horsepower respectively. At the rated power and



(Novo Engine Co.)

FIG. 229.—PERFORMANCE CURVES FOR NOVO ENGINE

speed of 40 and 50 B.hp. at 925 r.p.m., the specific gasoline rate is 0.64 lbs. for both engines. This indicates a low compression ratio, which is necessary if kerosene should be the fuel employed. The brake thermal efficiency on gasoline is about $2543 \div (20\,500 \times 0.64) = 0.194 = 19.4$ per cent.

The engines may be equipped to burn gasoline, gas, or kerosene, but the maximum power capacity on gas and kerosene is from 10 to 15 per cent less than with gasoline.

Many thousands of gasoline engines of the types represented by the Novo engines are in use for marine and land service. In a large measure they have been instrumental in eliminating much of the slavish toil of industry and in greatly increasing the output per worker.

226. Sterling Gasoline Engines of Large Power. A view of a Sterling gasoline engine working on the four-stroke cycle is shown in Fig. 230. These engines under the name of Viking II



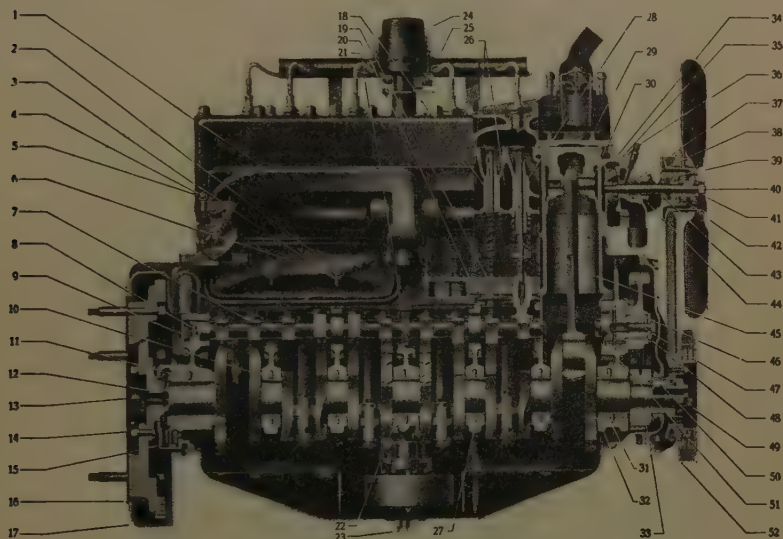
(Sterling Engine Co.)

FIG. 230.—565-HP. 8-CYLINDER VIKING II

are made up in six- and eight-cylinder units. In each unit the cylinders have an 8-inch bore and a 9-inch stroke. At 1200 r.p.m., they develop 425 and 565 brake horsepower respectively, and weight about 15 pounds per brake horsepower.

The Viking II engines are designed to compete with Diesel engines of the same power. The advantages claimed for them are light weight and correspondingly low first-cost made possible by the higher speeds and lower cylinder pressures. They have been placed in production only after a thorough analysis of the fuel situation, and the confirmed indications that the gasoline engine of relatively large power will continue in increasingly heavy demand. The development of cracking processes renders available an excellent grade of gasoline at a price which makes the operating expense of these engines economical. This line of engines is designed primarily to propel high-speed yachts and small pleasure craft, but they may be adapted for industrial purposes also.

227. Chrysler Six-cylinder Automobile Engine. Sectionalized views of the Chrysler six-cylinder Model "72" automobile engine for 1928 are shown in Figs 231 and 232. This engine is of the Otto four-cycle poppet-valve L-head type (the valves are installed in a chamber integral with and to one side of the top end of the cylinder, giving the cylinder somewhat the appearance of an inverted letter L).



(Chrysler Sales Corporation)

FIG. 231.—SECTIONAL VIEW OF AUTOMOBILE ENGINE

1, exhaust and intake manifold; 6, valve cover plate; 7, cam shaft; 10, crank-shaft bearing No. 6; 11, crank-shaft bearing No. 7 (rear); 15, rear oil drain passage; 16, flywheel; 17, flywheel housing; 18, valve guide; 21, distributor and oil-pump drive; 22, crank-shaft center bearing; 24, distributor; 28, poppet valve; 29, piston; 30, wrist pin; 32, main bearing No. 1; 33, crank-shaft sprocket; 34, cylinder flange for water pump; 35, water-pump impeller; 39, pump and fan-shaft bearing; 40, pump and fan-shaft; 45, connecting rod; 51, front bearing for cam shaft; 52, torque impulse neutralizer.

The cylinders are cast in one block, and are provided with a removable head. They have a $3\frac{1}{4}$ inch bore and provide for a 5 inch piston stroke giving a total piston displacement of 249 cubic inches per revolution of the crankshaft. The compression ratio is about 5.1 to 1. The maximum brake horsepower is about 75 at 3000 r.p.m.

The crankshaft is balanced dynamically and statically, and is mounted in seven bronze-backed babbitt-lined bearings. These bearings are of a special interchangeable type manufactured to such close limits that new ones may be installed without reaming, scraping, or burnishing. On account of the high-pressure lubricating system (25 to 30 pounds pressure) all engine bearings are

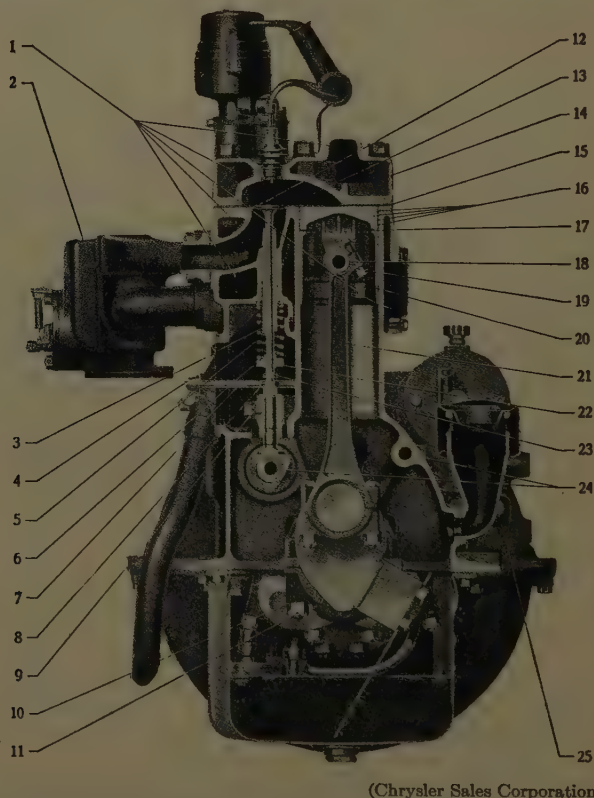


FIG. 232.—TRANSVERSE SECTION OF AUTOMOBILE ENGINE

1, water passages; 2, exhaust and intake manifolds; 3, valve guide; 4, valve spring; 5, tappet-adjusting screw; 6, valve tappet; 7, combustion chamber; 8, cylinder heat; 9, piston rings; 10, piston; 11, connecting rod; 12, valve; 13, combustion chamber; 14, cylinder heat; 15, piston rings; 16, piston; 17, connecting rod; 18, valve; 19, combustion chamber; 20, cylinder heat; 21, piston rings; 22, piston; 23, connecting rod; 24, oil passages; 25, oil-pressure relief-valve.

assembled with a clearance of 0.002 inch, thus allowing a film of oil under pressure to act as a cushion between the bearings and shaft.

A six-volt one-wire electrical system is employed; that is, the various units and the lighting circuit are grounded to the car

frame which serves as one side of the circuits. The several units comprising the system are the starting motor, generator, relay, ignition timer, distributor, high-voltage coil for the spark plugs, storage battery, lights, and horn.

Typical performance curves for automotive type engines obtained with the engine on the testing stand are shown in Fig. 233. The specific fuel-rate curve there shown is very flat over a wide range of speed. It is for full-throttle opening. At partial-throttle openings the fuel rate is somewhat higher.

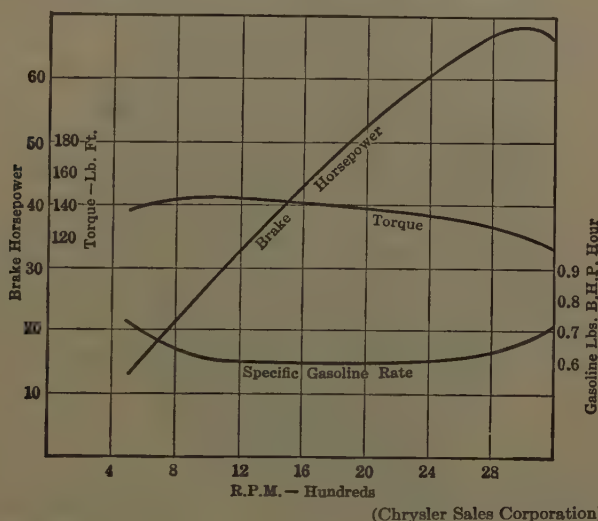
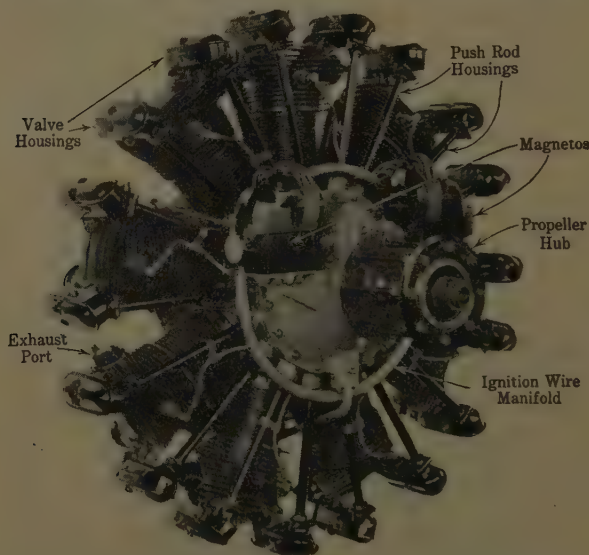


FIG. 233.—PERFORMANCE CURVES OF AUTOMOBILE ENGINE

Many auto engines with eight cylinders in-line are now being produced. Their design is similar to that of the six. Reduced vibration and greater flexibility are thus attained.

228. Airplane Engines. The outstanding characteristics demanded in airplane engines are absolute minimum weight per horsepower combined with extreme reliability and good fuel economy. Engines weighing about 0.5 lbs. per brake horsepower are available, but most of them weigh from 1.5 to 2.0 pounds. They are capable of 100 to 300 hours' continuous operation without being overhauled, and have a life of about 1000 hours in active service.

Views of the Wright "Whirlwind" engine,¹ used by Col. C. A. Lindbergh, Commander R. E. Byrd, and Clarence Chamberlin in their record-making flights, are shown in Figs. 234, 235, and 236. This engine is of the static nine-cylinder air-cooled, radial type operating on the four-stroke cycle. The bore and stroke are 4.5 inches and 5.5 inches respectively. The compression ratio is 5.2 : 1. The guaranteed brake horsepower at sea level is 200 hp.



(Wright Aeronautical Corporation)

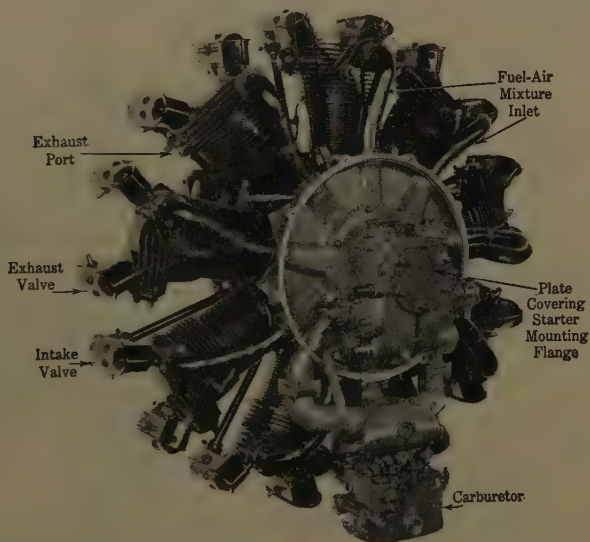
FIG. 234.—FRONT END "WHIRLWIND" AVIATION ENGINE

at 1800 r.p.m.; while maximum power obtained from the average engine at this speed is about 220 brake horsepower as indicated by the performance curves of Fig. 238. The average weight of the engine is about 500 pounds; hence its weight per brake horsepower is about 2.5 pounds. This is not the lightest engine available, but due to its design and weight it is one of the most durable and reliable of airplane engines of the present time. It has been proved that the operating expense of a modern air-cooled radial engine is much lower than for the water-cooled type; that its ultimate life is much longer; and that in the end the air-cooled engine is the more economical.

¹ For short historical sketch see *Mechanical Engineering*, Mar., 1929, p. 186.

The cylinder is made up of a separate head and barrel. The head with its integral fins is cast in aluminum alloy. It is screwed and shrunk on the forged chrome-nickel steel barrel having integral fins.

The combustion chamber is hemispherical with the valve stems inclined to the cylinder axis at an angle of 35 degrees. This form produces maximum pressure from the explosion. There are two spark plugs per cylinder located diametrically opposite each other. The cylinder head contains two ports, the axes of which



(Wright Aeronautical Corporation)

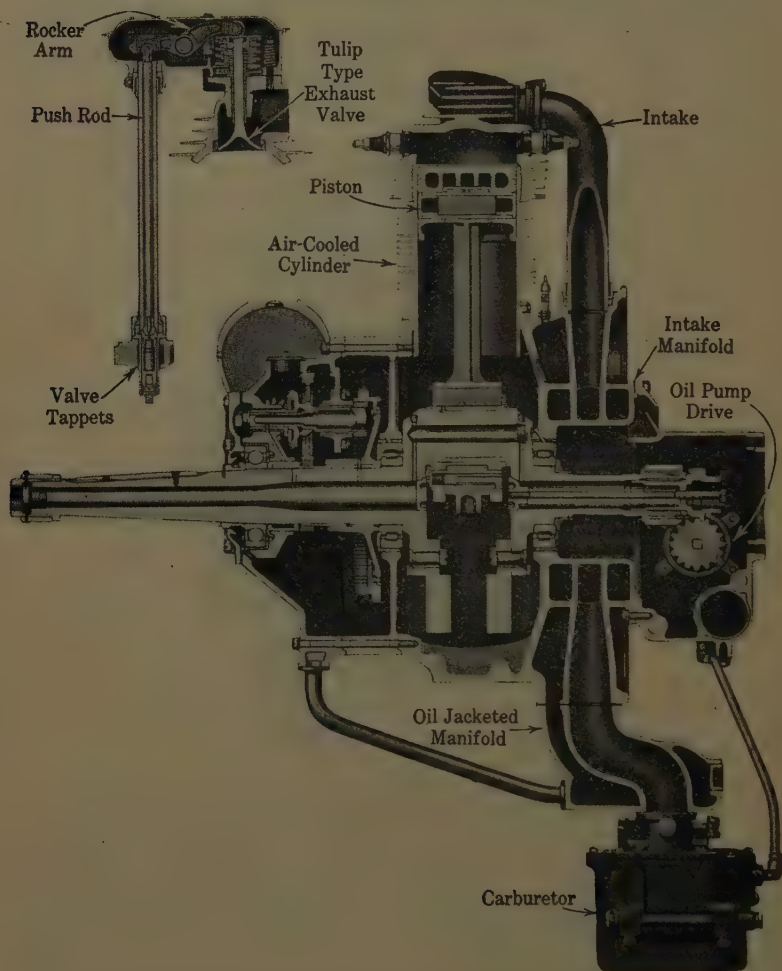
FIG. 235.—REAR VIEW "WHIRLWIND" AVIATION ENGINE

are at right angles to each other. The port facing aft is for the attachment of the intake pipe, and the exhaust port is at the side of the cylinder.

The crankshaft machined from a chrome-nickel steel forging is of the single-throw counter-balanced type. It is hollow throughout its length, permitting the distribution of lubricant under pressure to all moving engine parts. The shaft is carried by four bearings: a ball thrust-bearing, two main roller-bearings, and a plain bearing in the rear crankcase section where the oil is ad-

mitted. Gears which drive the valve cam and other accessories are attached to the crankshaft.

The crankcase is built up of five aluminum-alloy castings se-



(Wright Aeronautical Corporation)

FIG. 236.—SECTION THROUGH WHIRLWIND AVIATION ENGINE

cured together by studs and nuts. The main section is provided with flange seats for attaching the nine cylinders, and it also carries the flange by which the engine is attached to the plane.

The connecting rods consist of one master rod and eight link rods (Fig. 237) machined from chrome-nickel steel forgings.

The pistons are made of cast-aluminum alloy with the inside of the head heavily ribbed to give increased strength and cooling surface.

The valves are of the tulip type, and are machined from tungsten steel, the exhaust valves being salt-cooled. The valves are operated by an 8-lobed cam ring containing an internal gear. The cam ring is riveted to an aluminum hub riding on the sleeve portion of the cam drive gear which is keyed to the crankshaft.

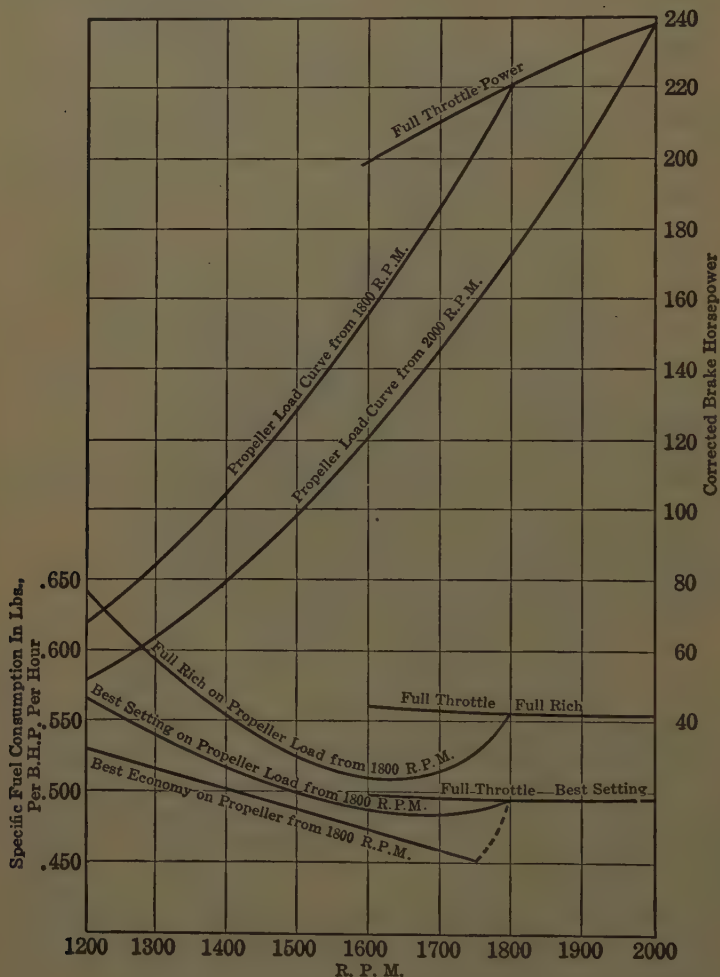


(Wright Aeronautical Corporation)

FIG. 237.—CONNECTING-ROD ASSEMBLY OF THE "WHIRLWIND" ENGINE

The cam followers are of the roller type operating in forged-steel bushings. The upper ends of the followers contain hardened-steel sockets into which the balls on the lower ends of the push rods fit. The push rods are of nickel-steel tubing with hardened-steel balls pressed into both ends. The upper balls fit into hardened-steel cups which are a loose fit inside the clearance-adjusting screws on the outer ends of the rocker arms. The valve gear is enclosed in rocker-arm housings which form the support for the rocker-arm pins. Pressed-steel covers close the rocker-arm housings, and the push rods are enclosed in steel tubes.

Ignition is accomplished by means of two Scintilla high-tension magnetos mounted on the front end of the crankcase. Each magneto is connected to all the cylinders, thus furnishing a complete dual-ignition system.



(Wright Aeronautical Corporation)

FIG. 238.—PERFORMANCE CURVES FOR "WHIRLWIND" ENGINE

The engine is equipped with a Stromberg carburetor of the three-barreled type with a mechanical economizer. It is attached to an oil-jacketed manifold sump fastened to the bottom

of the central section of the crankcase which contains three annular passages serving as intake manifolds to each group of three cylinders. Fuel enters each cylinder through steel intake tubes which connect them to the proper manifold.

Performance curves of the Whirlwind engine are shown in Fig. 238. The curve marked "Full Throttle Power" shows the maximum power of the average engine at the speeds indicated with full-rich setting of the mixture control. Individual engines may vary 2 or 3 per cent above or below the curve values.

At the full-rich setting of the mixture control, the specific fuel rate at full-throttle opening is indicated by the curve marked "Full Throttle Full Rich." The specific fuel-rate at full-throttle opening with the leanest mixture on which the engine will turn over at a maximum of 2000 r.p.m. is indicated by the curve marked "Full Throttle Best Setting." Hence, by use of the mixture control, at full throttle and at 1800 r.p.m., the specific fuel rate may be varied from 0.555 to 0.495 lbs. per hp. per hr. without loss in engine speed.

An engine which, with its propeller, will just turn 1800 r.p.m. at full-throttle opening, when throttled to any given speed will develop the power shown by the curve "Propeller Load Curve from 1800 r.p.m." If the full-throttle speed is not 1800 r.p.m., the propeller-load curve will start at the proper point on the "Full Throttle Power" curve, and will be similar in form to the "Propeller Load Curve from 1800 r.p.m."

The specific fuel rate along the propeller-load curve is given by the three curves marked "Full Rich on Propeller Load from 1800 r.p.m." The upper curve is for the full-rich mixture. The middle curve is obtained when the mixture control is set to give the leanest mixture at which the engine will just turn over a maximum of 2000 r.p.m., and left in that position as the engine is throttled down to lower speeds. The bottom curve is obtained when the mixture is the leanest on which the engine will operate smoothly without missing.

The maximum r.p.m. on the fuel-consumption curve must coincide with the full-throttle r.p.m. on the "Full Throttle Power" curve, and it must lie on the full-throttle specific-rate curve.

From this point the specific fuel-consumption curves for partial throttle opening will be similar in form to the curves drawn from 1800 r.p.m.

Air cooling together with the more advantageous form of the radial-type engine have permitted a material reduction in the span and weight of the airplane structure. The largest seaplanes equipped with Liberty V-type water-cooled engines weigh about 11,000 pounds complete. Planes of equal speed and carrying power but equipped with radial-type air-cooled engines weigh only about 7,000 pounds. This constitutes a very decided advantage in favor of radial air-cooled engines. But the vertical multicylinder water-cooled design is employed extensively in modern stationary and marine-type engines for both large and small powers.

229. Gas Engines. Engines which burn gaseous fuels only invariably operate on the Otto cycle. However, they may be of the four-stroke or of the two-stroke variety. The chief precaution is to see that the clearance volume and piston displacement are adjusted to give a compression pressure as high as possible without danger of producing serious preignition for the particular gaseous fuel to be used. But this precaution must be observed for all fuels on Otto-cycle engines. The compression pressures which have proven satisfactory with different gaseous fuels are given in Table 69.

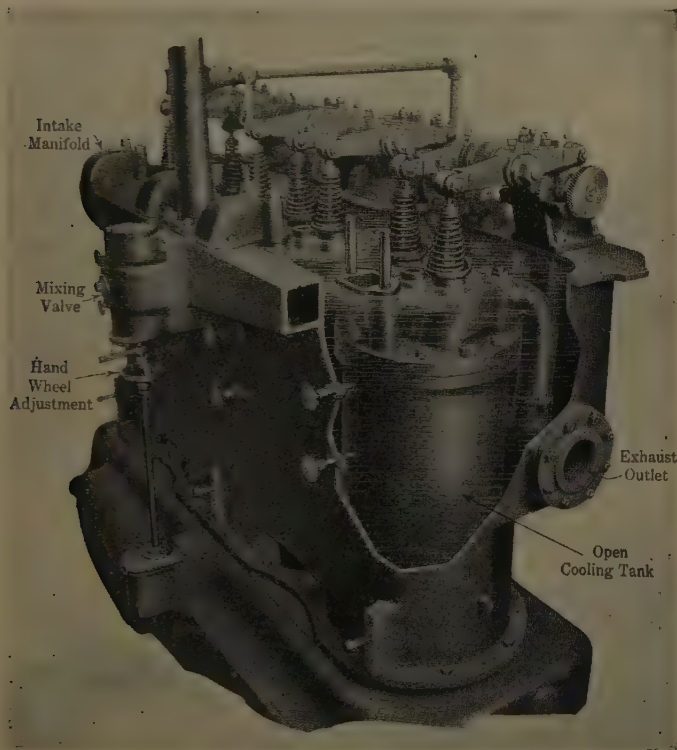
TABLE 69
END-COMPRESSION PRESSURES IN GAS ENGINES

FUEL	COMPRESSION PRESSURE LBS. PER SQ. IN. ABS.	FUEL	COMPRESSION PRESSURE LBS. PER SQ. IN. ABS.
Blast-furnace gas.	140-200	Natural gas.	100-150
Producer gas.	110-175	Illuminating gas.	90-125
Coke-oven gas.	115-150		

Gas engines are designed with one or several cylinders arranged vertically or horizontally. The horizontal cylinders may be placed in tandem or in twin-tandem arrangement. The earlier

large slow-speed internal-combustion engines (large in power and large in dimensions and weight) were developed to use gaseous fuels and were similar in form to those shown by Figs. 242.

230. Bruce-Macbeth Four-Cylinder Gas Engine. One form of modern multicylinder gas engine is illustrated in Figs. 239 and 240. The special feature of this engine is the open hopper

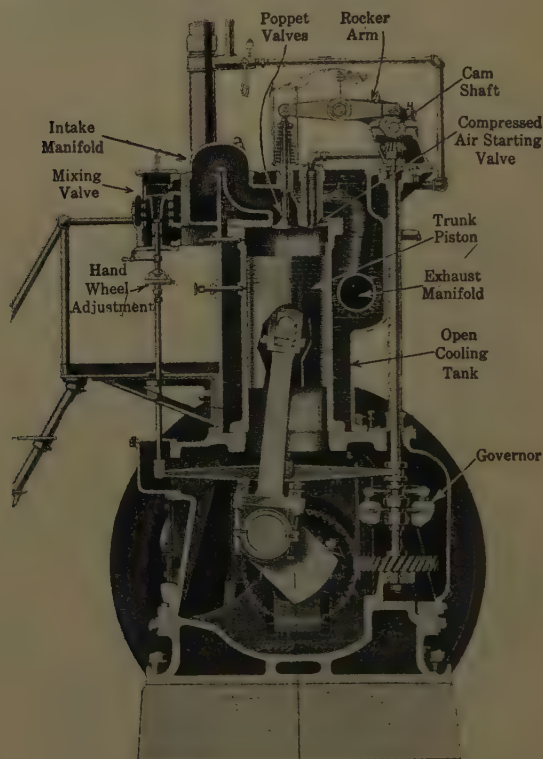


(Bruce-Macbeth Engine Co.)

FIG. 239.—FOUR-CYLINDER GAS ENGINE

which surrounds the cylinders in place of the more conventional water-jacket. This arrangement makes possible a simpler cylinder which is cheaper to manufacture, and less liable to crack from heat strain. The cylinder heads are also simple castings, and the valves are surrounded by water, cooling them most effectively.

The gas supply is controlled by a governor-operated throttle-valve which varies the richness of the mixture of air and gas in accordance with the load demand. The engine speed may be varied by means of a hand wheel which permits adjustment of the mixing valve while the engine is running.



(Bruce-Macbeth Engine Co.)

FIG. 240.—FOUR-CYLINDER GAS ENGINE

The valves are in the cylinder heads, and are operated by an overhead cam shaft which is driven through a vertical shaft geared to the main crankshaft. The centrifugal governor is mounted on the vertical auxiliary shaft. It is enclosed in the crankcase and is lubricated by the splash system.

The cast-iron pistons are of extra long trunk pattern carrying four rings especially ground to give uniform pressure against the cylinder walls.

Two complete high-tension magneto-ignition systems are provided, either of which will start and operate the engine. Hence uninterrupted operation of the engine is assured so far as ignition is concerned.

The engine is started by admitting compressed air to each cylinder through special starting valves operating by starting cams. When the air supply is shut off, the starting-valve stems drop automatically into the non-operative position; thus these valves and their mechanism are in operation only during the brief starting period. The approximate heat rate for a 100-horsepower Bruce-Macbeth engine is shown by the curve of Fig. 241. The B.t.u. required to develop one brake horsepower at various percentages of full load are indicated. At 100 per cent load, 10,000 B.t.u. are required per brake horsepower per hour. Since one horsepower-hour is the equivalent of 2543 B.t.u., the brake thermal efficiency of this engine at full load is $2543 \div 10\,000 = 0.2543$ or 25.43 per cent.

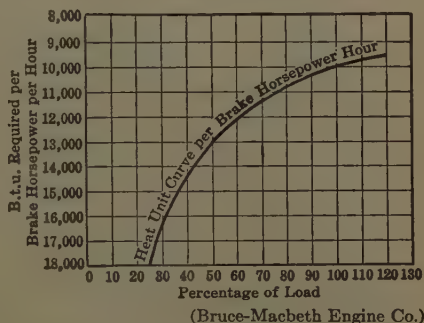


FIG. 241.—HEAT-RATE CURVE FOR GAS ENGINE
(Bruce-Macbeth Engine Co.)

231. Large Slow-Speed Gas Engines. Figure 242 illustrates a typical form of large heavy-duty gas engines which has become more or less standardized. Engines of this general design, which has been developed by long experience, are built in sizes ranging from 250 to 5000 horsepower. They are built to operate on natural, illuminating, coke-oven, or blast-furnace gas. They are of the horizontal four-stroke-cycle double-acting type in either two- or four-cylinder units. The cylinders are arranged in tandem for the two-cylinder unit, and in twin-tandem for the four-cylinder type. The latter are provided with two cranks set 90 degrees apart, and 8 power impulses are produced per revolution of the crankshaft.



FIG. 242.—2400 HP. DOUBLE-ACTING TANDEM PRODUCER-GAS ENGINE
(Mesta Machine Co.)

The rotative speed is low, usually between 75 and 150 r.p.m.; hence the construction must be massive throughout to care for the large forces acting, and to give long life to the units under the severe service conditions usually met.

Special care is required in designing the cylinders, cylinder heads, and pistons of these large units since they must be water-cooled. In the past much trouble from cracked cylinder castings and cylinder-head castings has been experienced due to temperature strains set up by unequal cooling of the parts. In the conventional design, the cylinder, with water jackets and air-gas mixing chamber and passages, is cast in one piece, special precautions being observed to reduce shrinkage strains to a minimum during the solidifying of the metal in the mold. A renewable liner of steel or special cast iron is used for the cylinder bore. Provision is also made to surround the valves of poppet type with water jackets for effective cooling. The water jackets are deep to insure adequate and uniform cooling. Hand holes are provided in the jackets to permit inspection and removal of scale when and if deposited.

The pistons and piston rods are hollow, to provide for water cooling. Because of their large size and weight, the pistons are carried on guides between the cylinders and at both ends.

The inlet valves are located on the top side at each end of the cylinder. The exhaust valves, one at each end, are on the under side. This arrangement facilitates the removal of any solid impurities carried by the gas. For blast-furnace or coke-oven gases, which carry more or less dust in suspension, this is a necessary provision. The exhaust valves are usually cored out for water cooling. The valves are operated by cams or eccentrics on a lay shaft driven through spiral gears from the main shaft.

Ignition may be accomplished by the low-tension make-and-break system, energy being supplied by a 110-volt direct-current generator driven from the lay shaft; or by the high-voltage jump-spark system with spark plugs. Ignition energy for starting the engine may be obtained from a storage battery or from the regular lighting circuit. Ignition systems are usually installed in duplicate to insure continuous engine operation in case of failure of one system.

The engine speed is controlled by a governor, usually of the centrifugal type, which controls the quantity or the quality of the air-gas mixture, or both. For the smaller engines, the governor actuates directly a butterfly or throttle valve which controls the richness of the mixture. But in the largest engines, an oil relay is generally required to furnish the large force required to move the large mixing or throttling valves.

These large engines are usually started by admitting compressed air on the working stroke to each end of one or two cylinders. The air is compressed by a small direct-driven compressor, and stored in a tank of suitable size. The air pressure employed is generally about 200 pounds per square inch, although lower pressures may be used, but larger storage tanks are then necessary.

The importance of small and medium-sized gas engines in the production of power is rapidly approaching the vanishing point. The rapid development of gasoline and Diesel engines is largely responsible for the decline. However, in favored localities where natural gas or blast-furnace gas is available in sufficient quantities, large gas engines are still most economical prime movers.

Test results of a 3000 horsepower twin-tandem blast-furnace gas engine are given in Table 70.

TABLE 70
TEST RESULTS FROM BLAST-FURNACE GAS ENGINE

R.p.m.	81.3
Bore of cylinder (inches).....	42
Piston stroke (inches).....	60
Full-load rating (kilowatts).....	2200
Average load (kilowatts).....	2190
Average brake horsepower.....	3090
Efficiency of generator per cent.....	95
Cubic feet of gas per D.hp. per hr.....	94.2
B.t.u. per cu. ft. of gas (higher).....	86.6
B.t.u. per D.hp. per hr.....	8170
Brake thermal efficiency per cent.....	31.2
Total thermal efficiency, power at switchboard.....	29.6

232. Diesel Engines—General Considerations. Of all the varieties of internal-combustion engines, the Diesel has become most important in power generation outside the automotive and

airplane fields, and it promises to grow in importance as time passes. Eventually it may become the chief prime mover even in these automotive fields. American, British, and German-built Diesel engines made successful airplane flights in 1929. The American engine built by the Packard Motor Co. is a 9-cylinder radial type developing 200 horsepower at 1550 r.p.m. and weighing about 2.25 pounds per horsepower.¹ In thirty years of intensive development it has reached a degree of perfection and reliability hardly attained by the steam engine even after two hundred years of development. The chief factor responsible for this rapid perfection of the Diesel is its extremely economical use of fuel. Also the fuel economy of the smallest unit is nearly as good as that of the largest, as will be seen from the performance data given on succeeding pages. The Diesel possesses other features also which make it superior to the other internal-combustion motors. It burns heavy liquid fuels of high boiling point, which are much cheaper than gasoline. It requires no carburetor. It requires no troublesome ignition devices which are liable to become disarranged.

On the other hand, there are several inherent features which prevent the Diesel from being the perfect motor. The large forces operating intermittently require a massive design. The maximum rotative speed is low. Up to the present time, it has been impossible to operate them at speeds much above 1500 r.p.m.² The vast majority are driven at speeds between 90 and 300 r.p.m. These two factors (large forces and low speeds) combine to give necessarily a very heavy engine per unit of capacity compared to gasoline engines. It has not been feasible to build Diesel engines of small power (one to ten horsepower per cylinder) because of the difficulty of properly measuring out the very small volumes of liquid fuel to be injected per explosion. Ten horsepower per cylinder is about the smallest available. The smallest size built by most firms develops about 30 horsepower per cylinder.

Suitable nozzles whose development began as early as 1890 employing only high pressure (2000 to 5000 lbs. per sq. in.) to

¹ See Mechanical Engineering, Jan., 1930, p. 38.

² See Mechanical Engineering, Aug., 1929, p. 571.

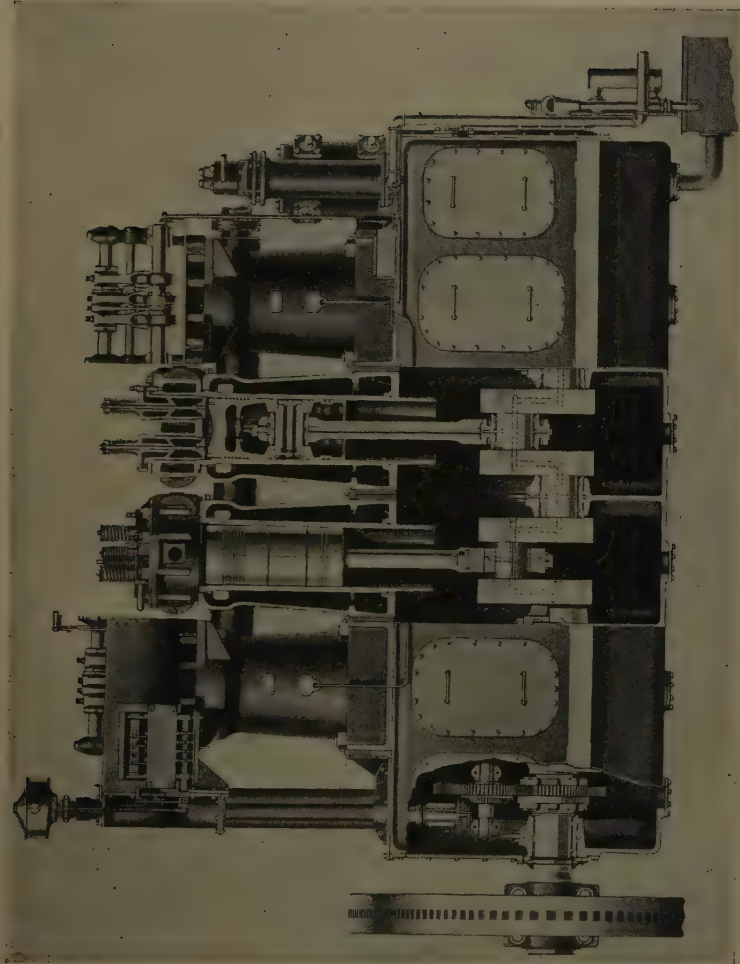
spray the fuel into the cylinder are now available and produce satisfactory results for the smaller engines (from 30 to 1000 horsepower units).¹ This method of spraying the fuel is given various names as hydraulic, mechanical, airless, solid, or direct injection. For engines larger than about 1000 horsepower, however, the use of compressed air seems to be the only satisfactory method yet developed for spraying the larger quantities of fuel into cylinders developing large power. Solid injection eliminates the high-pressure air compressor, which is costly, somewhat dangerous, complicated, and a source of much trouble. Moreover, with airless injection, the useful horsepower developed is from 5 to 10 per cent greater for the same cylinder dimensions and engine speed. As development of solid-injection nozzles proceeds, they will doubtless become available for much larger engines than at present.

233. Worthington Air-Injection Diesel Engine. Sectionalized views of a Worthington air-injection Diesel is shown in Figs. 243 and 244. The engine has four vertical cylinders each 17" $\times$ 25" which develop 100 horsepower per cylinder at 200 r.p.m. on the four-stroke cycle. The design of this engine is more or less typical of all four-cycle air-injection Diesels.

The vertical arrangement of cylinders is advantageous and has become standard practice because any number of cylinders from 1 to 12 may be placed in line, giving a large range in possible engine powers with a minimum number of different parts to manufacture. (See Table 73, page 572.) All the important forces act vertically or nearly so, thus insuring uniformity of wear in the main bearings. The vertical arrangement of cylinders also gives an engine of large capacity for a given weight and floor space required, all of which are important considerations.

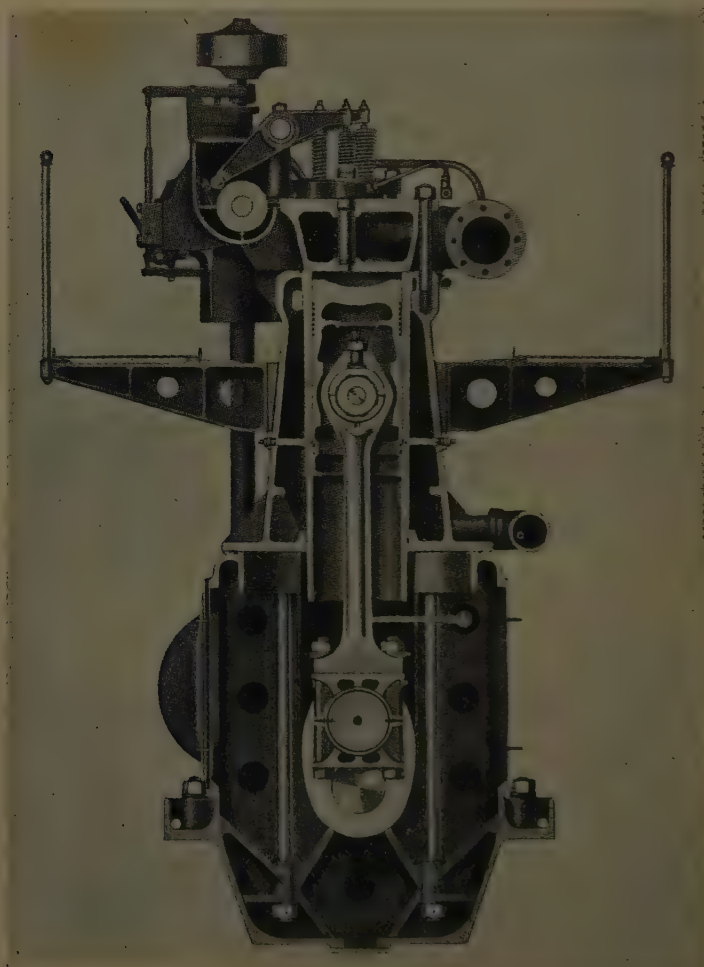
The cylinders of the Worthington engine are fastened to the base by long tension bolts. The cylinders are cast separately and each one is fitted with a removable cast-iron liner. The inlet, exhaust, fuel, and air-starting valves, in removable cages, are located in the water-cooled cylinder head.

¹ For a brief historical sketch on hydraulic injection, see Transactions A.S.M.E., vol. 45, 1923, p. 471.



(Worthington Pump and Machinery Corporation)
FIG. 243.—17 X 24 AIR-INJECTION DIESEL ENGINE

The overhead cam shaft and valve gear are driven from the main crankshaft by a combination of spur and bevel gears through a vertical shaft which also carries the centrifugal governor seen at the left in Fig. 243.



(Worthington Pump and Machinery Corporation)

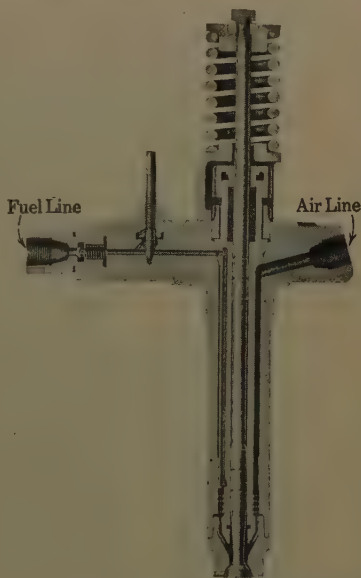
FIG. 244.—17 X 24 AIR-INJECTION DIESEL ENGINE

The fuel spray valve is one of the most important parts of a Diesel engine. Its construction and proper functioning are responsible, very largely, for correct and complete combustion

which is necessary for the economical generation of power. It is called upon to properly atomize all grades of fuel oil from the light distillates to heavy asphaltic residues. The Worthington spray-valve assembly (Fig. 245) is self-contained and removable from the cylinder head as a unit. It consists of a fuel-line connection containing a check valve and a priming valve; also an air passage connected to the high-pressure air line. The valve and stem are made of hardened stainless steel seating on a hardened surface, located some distance from the tip of the nozzle. The valve opens inwardly into the cylinder. This arrangement protects the valve and its seat from the heat of the combustion. The enlarged end of the valve aids in spreading the atomized fuel in a cone-shaped spray as it enters the cylinder.

The long trunk pistons have concave heads which tend to give a hemispherical shape to the combustion chambers. For Diesel engine sizes up to about 100 horsepower per cylinder, it is not necessary to water-cool the pistons, nor to equip the engines with crossheads. However, both are required for the larger cylinders.

The fuel pump is mounted on the front of the cam-shaft trough and below the Massey-Johns type governor (Fig. 243). The pump has a separate plunger for each power cylinder. A partially cut-away view of a 6-cylinder pump is shown in Fig. 246. The body of the pump consists of a single forging bolted to the cam-shaft trough. The plungers are of hardened steel, ground and lapped to a perfect fit in each pump barrel. No packing of any kind is necessary. The plungers are operated by cams



(Worthington Pump and Machinery Corporation)

FIG. 245.—AIR-SPRAY FUEL VALVE

mounted on the main cam shaft. The quantity of fuel delivered by each plunger stroke, and consequently the engine speed and power, is controlled by holding the suction valve open during part of the delivery stroke of the plunger. This is accomplished by a push rod under each suction valve mechanically operated by, and interlocking with, the plunger lever. The levers operating the push rods work on an eccentrically mounted shaft, the

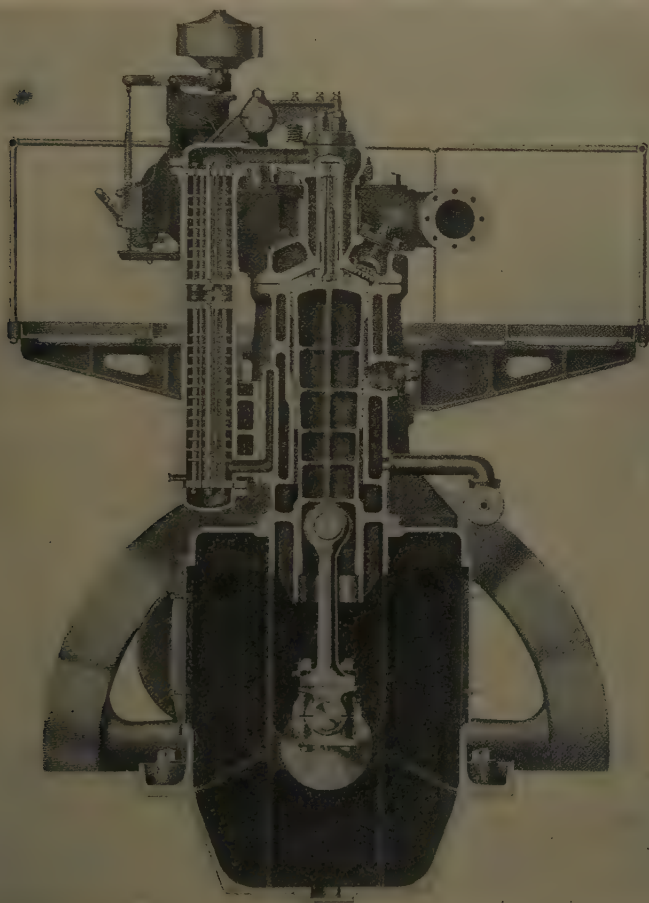


FIG. 246.—DIESEL FUEL PUMP

angular position of which is controlled by the governor. The action of the governor shifts the eccentric, thus varying the point in the discharge stroke at which the suction valves close. The control of the fuel pump can be taken away from the governor, and the amount of fuel delivered to each spray nozzle can be fixed by setting a hand throttle. Also each plunger can be hand-operated for filling the fuel lines before the engine is started. In

starting a cold engine, it is usual to use kerosene or some light distillate for fuel until the engine is partially warmed up.

A single-acting three-stage tandem air compressor (seen at the right in Fig. 243, and shown in section in Fig. 247) which



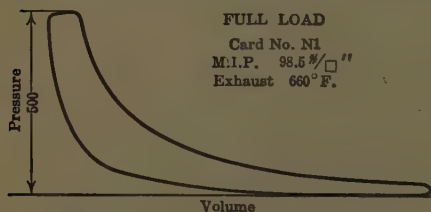
(Worthington Pump and Machinery Corporation)

FIG. 247.—THREE-STAGE AIR COMPRESSOR

supplies the high-pressure atomizing air, and the relatively low-pressure starting air is mounted on one end of the engine frame. It is driven by a smaller crankshaft extension from the main shaft. The air-compressor cylinders are cast with double

walls to permit water-cooling. The air is also cooled between stages by being passed through nests of intercooler tubes. The final pressure of the atomizing air is controlled by bleeding from the first-stage intercooler. This method is advantageous because it also removes a large percentage of the moisture and impurities from the air. In addition, traps for removal of any additional moisture and impurities are located at the outlet of the other two coolers. It is claimed that this method of pressure regulation gives a lower final discharge temperature than would be possible if the regulation were secured by throttling the first-stage suction. Air-storage tanks or bottles are provided, one high-pressure (800 to 1000 lbs. pressure) for the atomizing air, and usually two low-pressure (about 200 lbs. per sq. in.) for starting air.

If the store of high-pressure air is lost due to accident or long shut down, it is unnecessary to recharge the bottle from an out-



(Worthington Pump and Machinery Corporation)

FIG. 248.—INDICATOR CARD FROM 17 X 25 DIESEL ENGINE

side source. The bottle head is equipped with a by-pass valve which passes the atomizing air direct from the last intercooler to the air manifold. A few turns of the engine on the low-pressure starting air is sufficient to build up the atomizing-air

pressure to a value which properly atomizes the fuel. By allowing the bleeder valve in the first-stage intercooler to remain closed for a short time, sufficient excess air will be delivered to recharge the high-pressure bottle.

A full-load indicator card reproduced from the original is shown in Fig. 248. This is taken from a Worthington four-cycle air-injection Diesel engine. Since the indicated horsepower equals (n being the number of power impulses)

$$\text{I.h.p.} = \frac{\text{plan}}{33\,000} = \frac{98.5 \times 25 \times 226 \times 100}{33\,000 \times 12} = 140.5$$

The mechanical efficiency of this engine is probably between

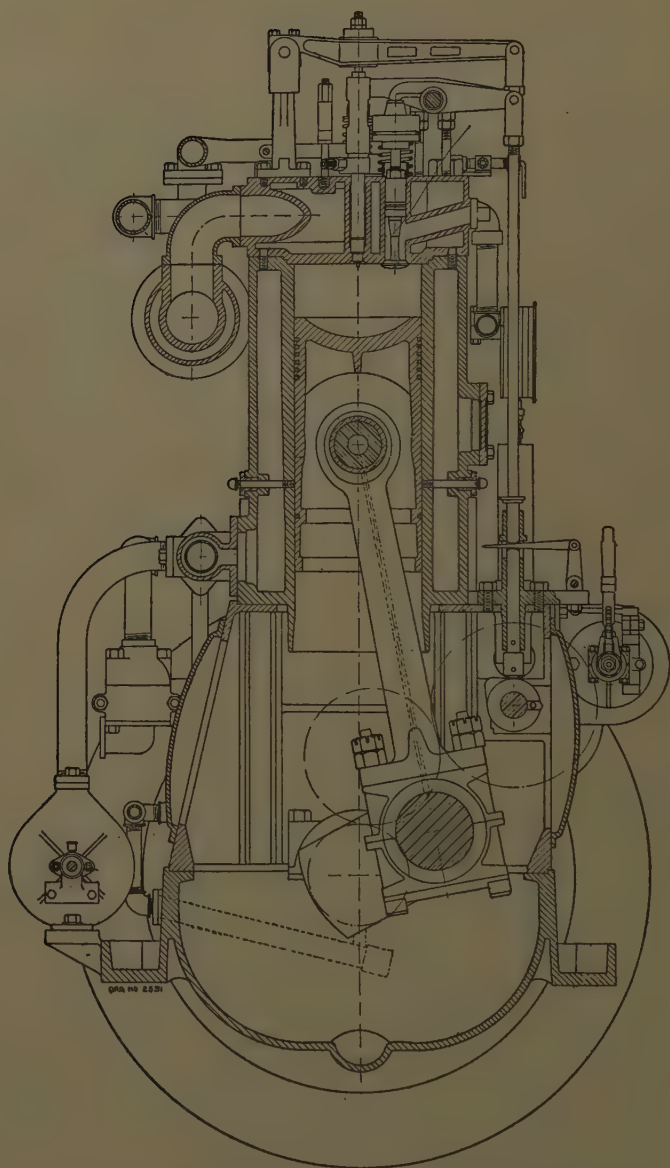
75 and 80 per cent; thus the developed horsepower per cylinder equals $140.5 \times 0.75 = 105.5$. Cylinder arrangements and fuel rates are given in Table 73, page 572 herein.

234. Atlas-Imperial Airless-Injection Diesel Engine. The general structural features of Atlas-Imperial engines are shown in Figs. 249 and 250. These engines were among the first to be built in America in relatively small powers for marine as well as land service, a number of them being put into service in 1915.

They are built in sizes ranging from 10 horsepower to 125 horsepower per cylinder, and for rated speeds from 275 to 750 r.p.m., the smaller engines being operated at the higher speeds. From 2 to 8 vertical cylinders are assembled in line developing from 20 to 1000 horsepower. These engines have moderately high rotative speeds in order to keep the weights at a low value, an important factor for marine service. The weight varies from 100 to about 200 lbs. (see Table 71) per horsepower.

The engines may be equipped with a geared reversing clutch (usual for the smaller powers) or with direct-reverse gear. The reversal of the engine is accomplished by manipulating a single control lever which shifts the valve operation from the go-ahead cams to the back-up cams, thus changing the timing of the valves. The time required to change from full-speed ahead to full reverse is from 3 to 6 seconds. A cold engine using heavy fuel oil may be started by means of compressed air, and full load applied in from 4 to 10 seconds, complete combustion in all cylinders being realized.

Fuel injection is accomplished by means of the hydraulic pressure of the liquid fuel (2000 to 4000 lbs. per sq. in.) through a carefully designed spray valve shown in Fig. 251. All the advantages of the air-injection type of engine are obtained without the necessary complications of two- or three-stage high-pressure air compressor, intercoolers, and air-storage bottles. Certain dangers incidental to the storage and use of high-pressure air are also eliminated. The most important item, however, is the saving in power. The air compressor in the air-compressor type absorbs from 5 to 12 per cent of the I.h.p. of the engine, giving it a



(Atlas-Imperial Engine Co.)
FIG. 249.—AIRLESS-INJECTION MARINE DIESEL ENGINE

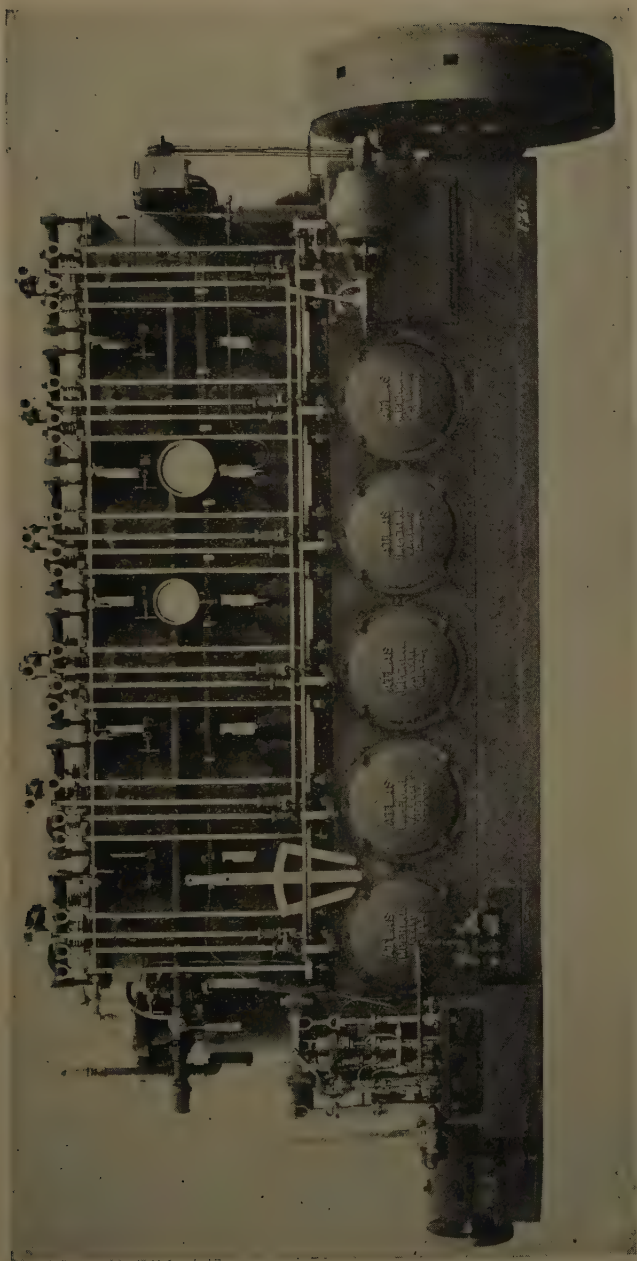


FIG. 250.—180-HP. DIRECT-REVERSIBLE AIRLESS-INJECTION MARINE DIESEL ENGINE
(Atlas-Imperial Engine Co.)

very low mechanical efficiency. The adiabatic expansion of the air blast also tends to cool the cylinder charge during fuel injection thus reducing the mean effective pressure. This action is especially noticeable at small fractional loads. However, each engine is equipped with a small air compressor driven by an eccentric on the main crankshaft; which air compressor furnishes the starting air at a pressure of about 200 pounds per sq. in.



(Atlas-Imperial Engine Co.)

FIG. 251.—AIRLESS-INJECTION VALVE

In 1926, nearly 60 per cent of all the horsepower built in stationary Diesel engines in this country utilized hydraulic (airless) injection. The trend in all countries is rapidly toward the use of hydraulic injection in the largest engines. In this country experimental engines which develop 1250 brake horsepower per cylinder have been built employing hydraulic injection.¹ In England a 2900-hp. four-cylinder two-cycle Doxford marine engine employing hydraulic injection has been operated successfully since 1926. (See Table 81, page 584 herein.)

The cylinders and cylinder heads of the Atlas engine are cast separately of a special semisteel mixture. Both are provided with ample water-cooling jackets, and are carefully designed to permit expansion and contraction due to heating and cooling without danger of rupturing. A relief valve is connected to each cylinder to safeguard it against excessive pressure. The overhead valves are operated by tappet rods from a horizontal cam shaft geared to the main shaft and enclosed in the crankcase.

Long-skirted pistons are employed, the cylinders being long enough to guide them throughout the stroke; hence crossheads and guides are unnecessary. The pistons are not water-cooled.

¹ See *Mechanical Engineering*, Jan., 1928, pp. 28-32.

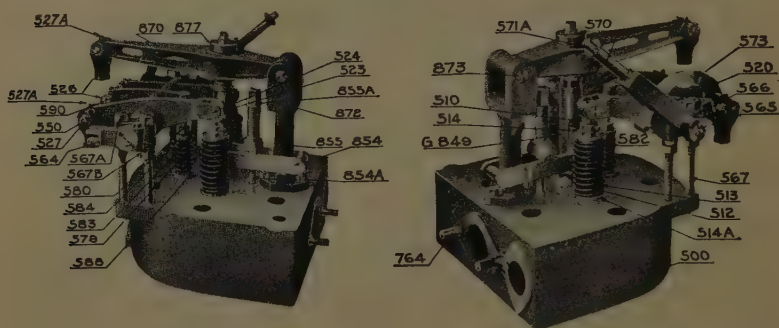
Experience has indicated that water-cooled pistons are not essential in engines developing less than about 125 horsepower per cylinder.

TABLE 71
DATA FROM ATLAS-IMPERIAL DIESEL ENGINES

RATED D.H.P. PER CYLINDER	CYLINDER DIMENSIONS, INCHES	RATED R.P.M.	CYLINDER COMBINATIONS	NET ENGINE WEIGHTS, LBS.
10-15	6.5 × 8.5	475-750	2, 3, & 6	2-cyl.-4 160 6-cyl.-8 500
16.67-20.0	7.5 × 10.5	400 & 450	3, 4, 5, & 6	3-cyl.-8 300 6-cyl.-12 000
21.67-23.3	8.5 × 12	325	3, 4, & 6	3-cyl.-11 100 6-cyl.-17 640
25-27.5	9 × 12	325	3, 4, & 6	3-cyl.-12 500 6-cyl.-18 200
30-31.25	9.5 × 13	325	3, 4, 5, & 6	4-cyl.-17 650 5-cyl.-20 500 6-cyl.-21 300
33.3-34.0	10 × 13	325	3, 4, 5, & 6	3-cyl.-16 000 4-cyl.-18 100 6-cyl.-22 300
41.25-41.7	11 × 15	275	3, 4, & 6	3-cyl.-20 940 4-cyl.-27 500 6-cyl.-36 500
45	11.5 × 15	275	3 & 4	3-cyl.-23 200 4-cyl.-31 000
50	12 × 16	275	6	43 000
66.67	14 × 18	250	6	61 000

Cylinder dimensions, rated power per cylinder, rated speeds, and weights of the regularly listed sizes of Atlas-Imperial engines are given in Table 71. Larger engines developing up to about 125 horsepower per cylinder, and with as many as eight cylinders per engine, may be obtained on special order. An examination of the

table shows that the engine weights vary from about 100 lbs. to about 208 lbs. per rated horsepower, but the most of them range between 150 and 170 pounds. The lighter and smaller engines operate at the higher speeds.



(Atlas-Imperial Engine Co.)

FIG. 252.—CYLINDER HEAD FOR AIRLESS-INJECTION DIESEL

500, cylinder head; 520, inlet rocker arm; 550, exhaust rocker arm; 565, rocker shaft; 570, air starting lever; 580, air starting valve; 590, air starting rocker arm; G849, spray valve complete; 854, spray valve clamp; 855, spray valve clamp bridge; 870, spray valve lifting arm; 872, spray valve rocker stand.

Test results of a 300 horsepower Atlas-Imperial engine at full load are given in Table 76, page 581. Comparison with other engine results indicate that the fuel rate of these small engines is as low as it is for the very largest Diesel engines.

235. The Two-Cycle Diesel Engine. As time has passed and experience has been accumulated in designing and operating Diesel engines, it has become generally recognized that the two-cycle style is the more suitable for engines of large size (above 500 horsepower) for both marine and land service. Their great advantage lies in the fact that one power impulse per cylinder is obtained each revolution in a single-acting engine. Thus, for an engine of fixed cylinder dimensions, speed, and weight, nearly twice the power is obtained. The power available is not exactly twice (the factor is from 1.7 to 1.9) that of the four-cycle type, largely because the scavenging is not so complete. A certain proportion of the burned gases remain in the cylinder to dilute

the charge for the succeeding explosion, thus reducing the force of all explosions. The two-cycle style also permits the elimination from the cylinder head of the air intake and exhaust valves, which are a source of much trouble. The cylinder-head casting is then much simpler and may be cooled effectively, greatly reducing breakage caused by temperature stresses. There are other advantages which accrue from the smaller sizes and weights of the reciprocating parts of the two-cycle type. The vibrations and shocks whose intensity vary as the weight and as the square of the velocity of the moving parts are less for a given power and speed. Because of the more uniform turning moment and the shorter crankshaft required, the torsional stress is less, thus permitting the use of a smaller shaft. For a given power and piston speed, Lloyd's rules for marine engines permit the crankshaft of a four-cylinder two-cycle engine to be about 10 per cent smaller than for a six-cylinder four-cycle engine. The crankshaft of a six-cylinder two-cycle engine may be 30 per cent smaller than for a six-cylinder four-cycle engine of the same power. Smaller foundations are also possible for the two-cycle engine. However these advantages are compensated to some extent by the necessity of a low-pressure (2 to 10 lbs. per sq. in. gage) air compressor to supply the scavenging air. The somewhat more imperfect scavenging also tends to increase slightly the fuel consumption of the two-cycle engine.

The Sulzer brothers of Switzerland, in spite of their large business in building four-cycle Diesel engines, were early convinced of the advantages of the two-cycle style for large engines. They had the money and patience to carry out extensive experimental work on the two-cycle engine. They built an experimental two-cycle engine developing a maximum of 2000 hp. in a single cylinder, and subjected it to severe test conditions for a considerable period. The present success of the two-cycle multi-cylinder engine is largely the result of this experimentation.

236. The Busch-Sulzer Two-Cycle Diesel Engine. Views of the Busch-Sulzer two-cycle vertical Diesel engines, are shown in Figs. 253 and 254. These engines are built in four- and six-cylinder combinations which deliver from 500 to 7000 horsepower

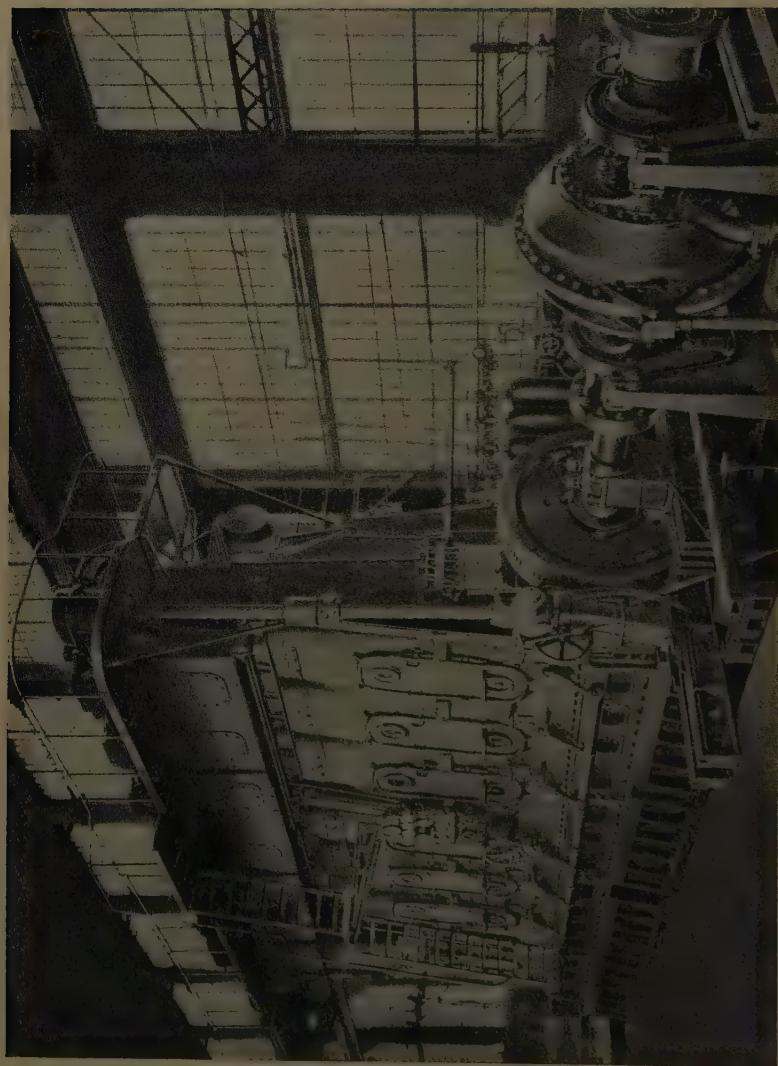
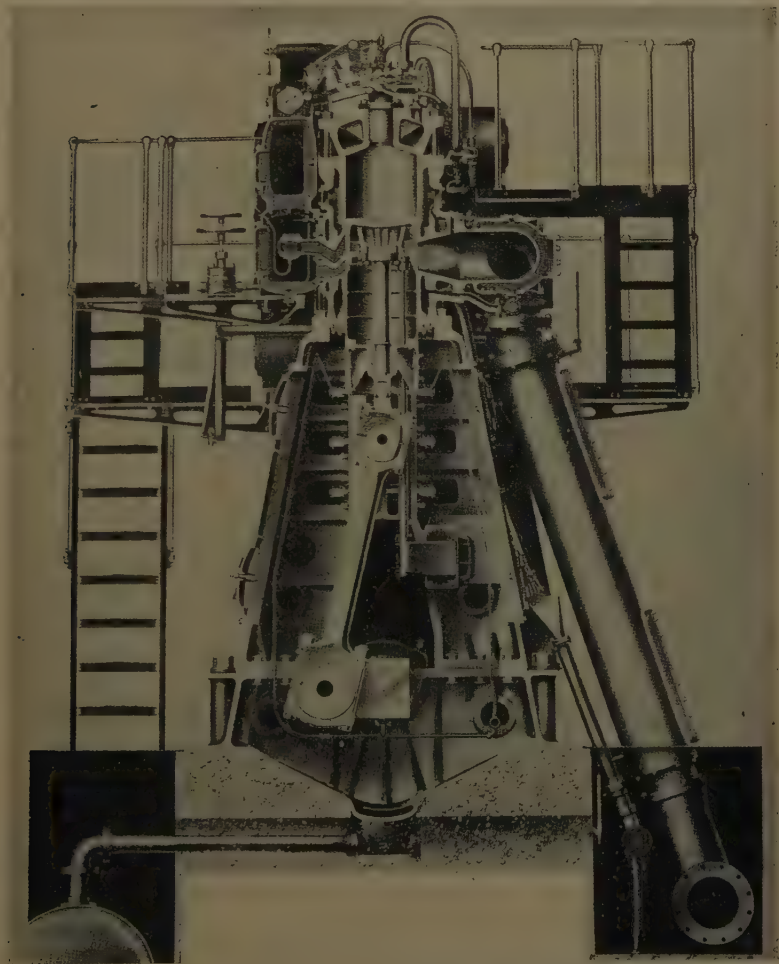


FIG. 253.—TWO-CYCLE 6-CYLINDER 30 X 52 3000-HP. DIESEL ENGINE
(Busch-Sulzer Bros.—Diesel Engine Co.)

at speeds which range from about 90 r.p.m. to about 250 r.p.m. The higher speeds are employed in the smaller engines. The engines are employed in both land and marine service. The chief



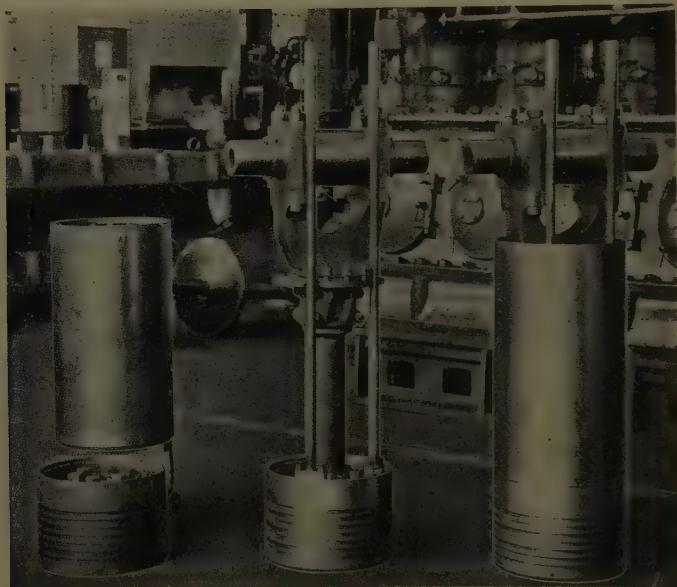
(Busch-Sulzer Bros.—Diesel Engine Co.)

FIG. 254.—TWO-CYCLE 3000 HP. DIESEL ENGINE

difference between them is that the marine engines are reversible. The reversing gear consists of a double set of valve cams. When the engine is reversed, an air-operated mechanism throws one set

of cams out of action and brings into action the other set. The main features of land and marine engines are similar in all respects.

The cylinders are cast in two parts, the outer shell which carries all the axial stresses, and the liner which constitutes the working barrel. The space between them provides for the circulation of cooling water. The jacket and liner are provided with openings front and back for the scavenging air and exhaust



(Busch-Sulzer Bros.—Diesel Engine Co.)

FIG. 255.—PISTON AND SKIRT FOR TWO-CYCLE DIESEL ENGINE

connections. The liner is made of hard close-grained cast iron. The central portion of the liner at the scavenging and exhaust ports is machined to fit a bored seat in the jacket, and is provided with packings to make water-tight and gas-tight joints around the ports. The upper end of the liner is provided with a groove into which the tongue of the cylinder head fits, making a gas-tight joint. The lower end of the liner is left free, allowing for expansion and contraction due to temperature changes.

The cylinder heads are of simple symmetrical design containing

only one central opening of relatively small diameter into which the fuel valve and starting valve are inserted in a single cage. Ample water-jacket spaces are provided. The combustion side of the head and the top face of the piston are concave, giving a combustion space which closely approaches the ideal spherical form.

The piston head proper is short (Fig. 255), merely being long enough to carry the piston rings, since it is guided by the cross-head. A long cylindrical skirt is attached to its lower end, whose only function is to cover the exhaust and scavenging air ports. The piston is water-cooled, the water being conducted into and from the piston by a system of telescoping tubes which are so arranged as to prevent leakage without the use of stuffing boxes.

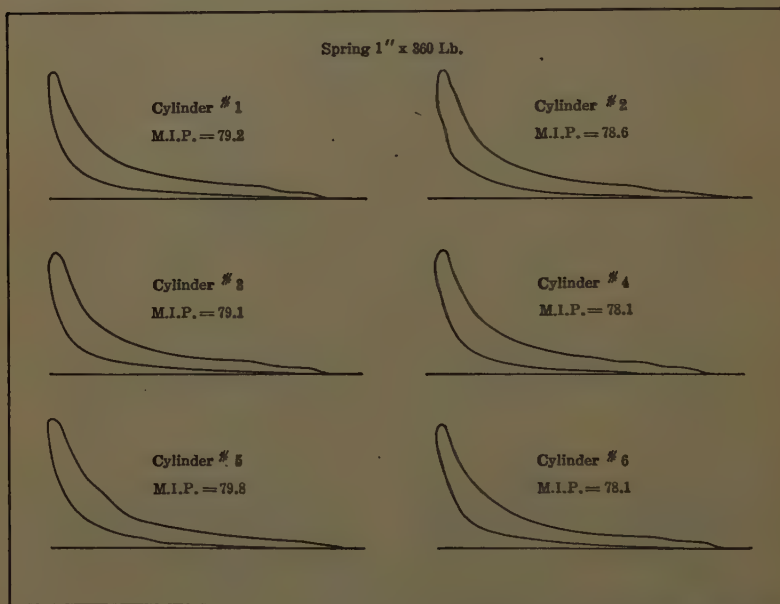
The crankshaft is in three sections joined together by heavy flange couplings. Each main section carries two or three cranks (two for a four-cylinder engine, and three for a six-cylinder engine). The third section carries the cranks for driving the scavenging and fuel-injection air compressors. The shaft is bored to permit inspection, and to afford passage for the lubricating oil. The design of the marine engine shafts meets Lloyd's specifications.

The bedplate is a box-girder type of structure heavily ribbed and flanged. It is built up of sections bolted rigidly together. It comprises an oil-collecting trough, and bridges which carry the bearings placed between each two cylinders; thus a four-cylinder engine has five, and a six-cylinder engine has seven main bearings.

The crankcase is of the enclosed type, and is oil-tight. It is built up in sections and contains the crosshead guides. It is bolted to the bedplate and the cylinder shells are bolted to its top flange. Cover plates, readily removable, are provided over each crank and connecting rod, making all parts inside the crankcase readily accessible.

The injection and starting air is compressed by a three-stage water-jacketed compressor provided with intercoolers and after-cooler. The compressor is mounted at one end of the bedplate. Safety valves are attached to each stage of the compressor to protect it against excessive pressure. The injection-air pressure

(800 to 1000 lbs. per sq. in.) is controlled by a special regulating device. The spray valves and the air-starting valves are operated by cams and rocker arms. There are two sets of cams—go-ahead and reverse—on the marine engines. The air-starting valves, when in operation, also relieve compression in the cylinders thus reducing to a minimum the pressure and volume of the starting air required. Land engines are usually equipped with air-starting valves on two cylinders only; while marine engines are so



(Busch-Sulzer Bros.—Diesel Engine Co.)

FIG. 256.—FULL-LOAD INDICATOR CARDS DURING OFFICIAL TEST OF 30 X 52 TWO-CYCLE ENGINE (NOT TO SCALE)

equipped on all cylinders to provide maximum maneuverability. When the engine is in operation only the fuel valves function.

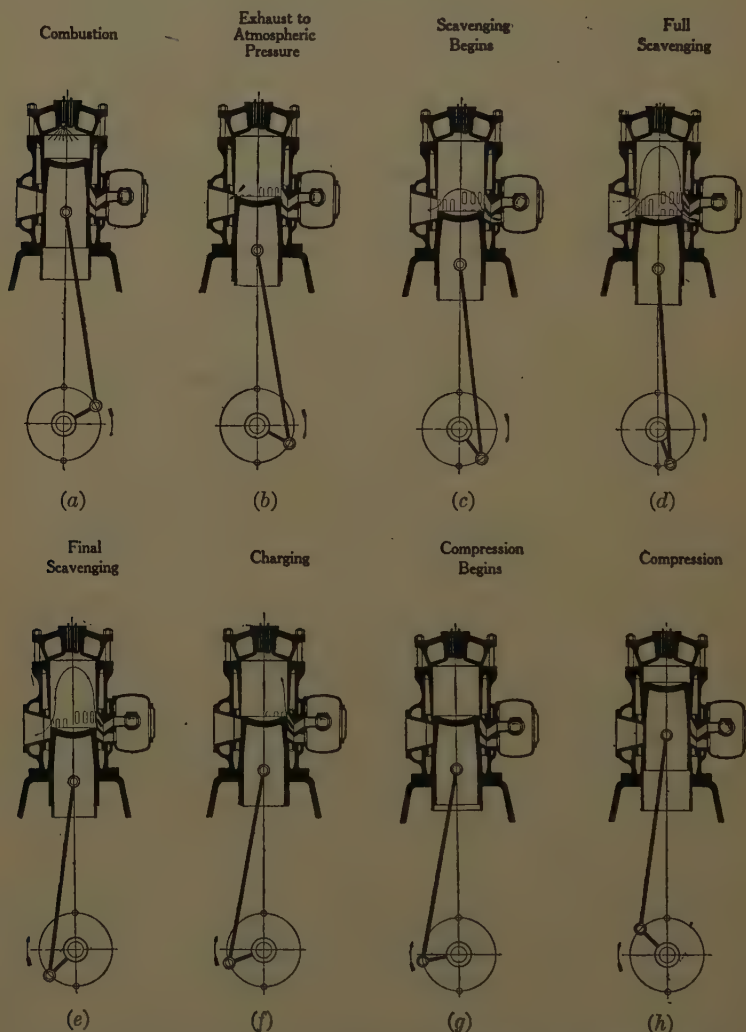
The scavenging air is provided at a pressure varying from 5 to 10 lbs. per sq. in. gage by a double-acting tandem compressor located between the injection-air compressor and the end cylinder of the engine. It is provided with crosshead and guides similar to those of the working cylinders. The scavenging air is delivered to a cast-iron receiver which extends along the front of the engine,

being bolted to the faces of the working-cylinder jackets. The receiver also serves as a support for the overhead cam shaft.

The scavenging-air and exhaust ports of the cylinders are opened and closed by the piston at the lower end of its stroke, uncovering them on the down stroke and covering them on the up stroke. The scavenging-air ports (left side of cylinder Fig. 254) are arranged in two tiers. The passage of the air through the upper tier is controlled also by a timed rotary valve driven from the vertical shaft which operates the cam shaft. The lower tier opens directly into the air receiver. This arrangement, it is claimed, secures nearly perfect scavenging, and locates the scavenging valve out of the direct range of the hot gases. A volume of scavenging air equal to about 1.6 times the cylinder volume is blown through both scavenging ports into the cylinder. The upper tier of scavenging-air ports remain open after the piston closes the exhaust ports; thus the cylinder is supercharged to the extent of the scavenging-air pressure.

A detailed study of the events as they occur during one cycle (two strokes) of the engine equipped with the special Busch-Sulzer scavenging and charging system is shown by Fig. 257. In Fig. 257 (a) fuel injection and combustion has just ceased. The piston covers all ports; the high-pressure gases are expanding along the curve $PV^{1.25} = K$ (approximately). In (b) the piston has just begun to uncover the exhaust ports. The upper tier of scavenging ports are uncovered, but the scavenging valve remains closed. The burned gases escape, reducing the pressure in the cylinder to atmospheric. At (c) the piston is uncovering the lower scavenging ports, and scavenging air begins to blow the burned gases out through the exhaust ports. At (d) the rotary scavenging valve has opened, and scavenging proceeds through both tiers of ports. At (e) the piston, beginning its return stroke, has covered the lower scavenging ports; the exhaust ports are partially covered; the scavenging valve remains open; and scavenging is nearly complete. At (f) the exhaust ports are closed; the scavenging valve remains open, charging the cylinder with pure air at a pressure of about 5 lbs. per sq. in. At (g) all ports are closed and compression begins. At (h) the piston ap-

proaches the top dead-center; the air pressure in the cylinder is approaching the maximum of about 500 or 600 lbs. per sq. in.; and the fuel valve is about ready to open.

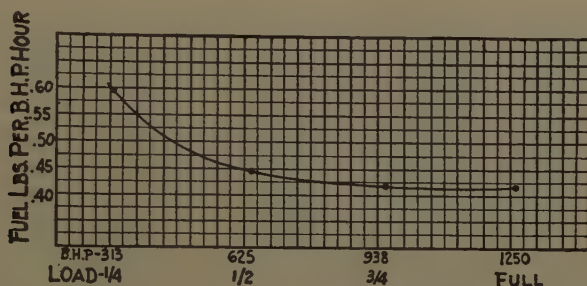


(Busch-Sulzer Bros.—Diesel Engine Co.)

FIG. 257.—EVENTS IN BUSCH-SULZER TWO-CYCLE DIESEL ENGINE

In Fig. 256 full-load indicator-cards are shown to a reduced scale. These cards were taken during a test on a six-cylinder,

30 × 52, 90 r.p.m., 3000-brake horsepower, single-acting, two-cycle, Busch-Sulzer, Diesel engine. The small variation in the work done in the different cylinders, as shown by the mean indicated pressure, is worthy of notice. A summary of the test results is presented in Table 72. This test was a 30-day acceptance test conducted by the U. S. Shipping Board in March and April, 1926.



(Busch-Sulzer Bros.—Diesel Engine Co.)

FIG. 258.—FUEL-RATE CURVE FOR 1250 HP. TWO-CYCLE DIESEL ENGINE

The specific fuel rate of a 1250-horsepower engine at various loads which is shown by the curve of Fig. 258 is characteristic of all Diesel engines.

Estimating the heating value of the fuel to be 18 500 B.t.u. per pound, the full-load thermal efficiency would be $2543 \div 18\,500 \times .42 = 0.328 = 32.8$ per cent. The very small variation

TABLE 72

30-DAY TEST OF BUSCH-SULZER TWO-CYCLE DIESEL ENGINE

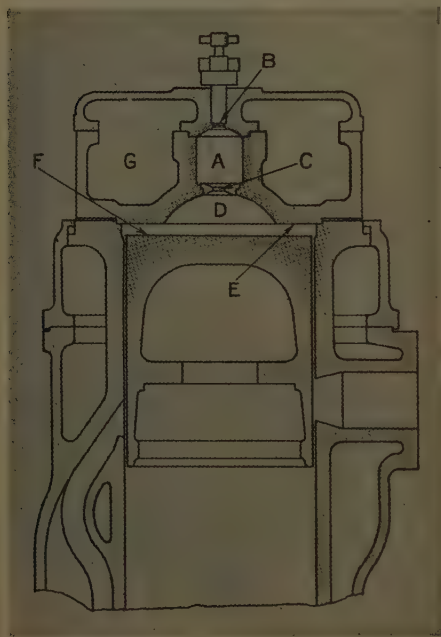
Average load, B.hp.	3012.07
Average engine speed, r.p.m.	89.25
Average mean indicated pressure, lbs. per sq. in.	81.53
Average brake mean effective pressure, lbs. per sq. in.	60.70
Average mechanical efficiency, per cent.	74.47
Average fuel consumption per 24 hrs, tons (2240 lbs.)	13.33
Fuel rate, lbs. per D.hp. per hr.	0.413
Heating value of fuel, higher B.t.u. per lb.	18 500
Brake thermal efficiency, per cent.	33.3
Average lubricating oil, gals. per 24 hrs.	21.5
Net weight of engine complete, without spares, starting tanks, oil, or water, thousands of lbs.	868.0
Net weight of engine per B.h.p., lbs.	289.3

in fuel rate between half-load and full-load is worthy of note as it is typical of all Diesel engines and a most desirable characteristic in all heat engines.

Four two-cycle marine Diesel engines, each developing 8000 horsepower in seven cylinders, having a bore of about 35.5 inches, and the rated speed being 87 r.p.m., similar to the Busch-Sulzer engines described above, have recently been completed.

237. The Worthington Two-Cycle Precombustion Engine.

The operation of the Worthington, two-cycle, semi-Diesel engine depends upon the fact that all fuel oils, no matter how heavy,



(Worthington Pump and Machinery Corporation)
FIG. 259.—CYLINDER AND HEAD FOR TWO-CYCLE PRECOMBUSTION ENGINE

contain small percentages of light components which ignite readily even when the compression pressure is as low as 250 lbs. per sq. in. A divided combustion chamber is employed as illustrated diagrammatically by Fig. 259. A precombustion chamber *A* is located in a two-part cylinder head. Its top end is closed by the cover which carries the spray nozzle *B*. A removable circular orifice *C* is screwed into the lower end of the chamber. The diameter of the orifice is so chosen as to control the pressure differentials, thus producing effective

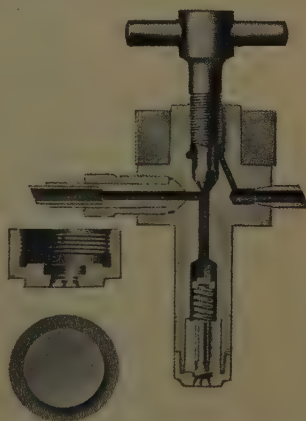
turbulence and spraying of the partially burned mixture of fuel and air during the injection period. A flat annular ring is machined on the bottom of the lower part of the cylinder head, and the top face of the piston is flat also. Hence, at the end of the upstroke the annular volume *F* is very small, and nearly all

the air is compressed into the dome-shaped space *D* directly beneath the orifice. This arrangement greatly aids in bringing the oil droplets into intimate contact with a maximum percentage of the compressed air as they are sprayed into the cylinder.

At the end of the upstroke, the compressed air fills the pre-combustion chamber *A* as well as the spaces *D* and *F*. The final pressure attains a maximum value of about 475 lbs. per sq. in. In chamber *A* the maximum is something less, perhaps 450 lbs. All the air has acquired a temperature, due to compression, somewhat above the ignition point of the lightest fractions in the fuel. The cylinder head is water-cooled, but the time required for the compression stroke is so short that the air is not cooled below the ignition temperature.

At about 15 degrees of crank angle ahead of dead center, fuel injection starts. Injection is accomplished by hydraulic pressure of the fuel alone. The fuel is directed in a downward direction toward the orifice *C* by the spray nozzle *B*, shown in detail in Fig. 260. Fuel injection continues until the engine governor opens a by-pass through which the remainder of the oil in the pump is returned to the suction side of the pump. Combustion begins immediately in chamber *A*, but due to the small quantity of air present, and because the oil is not broken up into extremely fine spray, a relatively small surface is exposed to the heated air, and only the lighter fractions burn. The oil spray and the gases of partial combustion tend to collect near the outlet of orifice *C*. Some of the air in the top and sides of the chamber *A* is not used during this first combustion period. This result is produced by properly proportioning the chamber dimensions and by the spray-nozzle action.

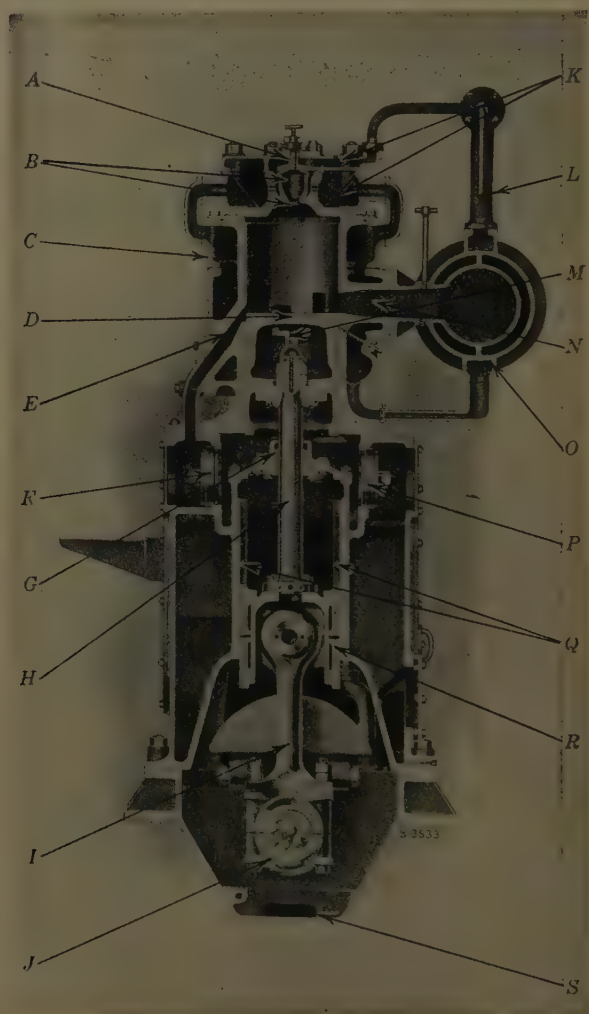
At about dead center, injection ceases, and the precombustion



(Worthington Pump and Machinery Corporation)

FIG. 260.—SOLID-INJECTION
SPRAY VALVE

period ends. During this period, the pressure in the precombustion chamber *A* has been raised by the partial combustion



(Worthington Pump and Machinery Corporation)

FIG. 261.—TWO-CYCLE SOLID-INJECTION PRECOMBUSTION ENGINE

from something less to a value somewhat greater than that in the cylinder; just how much greater depends upon the action of the spray nozzle, the proportions of the chamber, and the diameter

of the orifice. This increased pressure forces the mixture of burned gases and fuel spray through the orifice into the main cylinder at high velocity. This outflow is further accelerated by the downward movement of the piston. This completes atomization of the fuel, and a rapid and complete combustion is obtained in the cylinder.

Many precombustion engines, often called semi-Diesel engines, have been brought out and give fairly satisfactory service. They vary only in the detailed method of injecting the fuel into the cylinder. Their chief advantage lies in the fact that the precombustion arrangement gives a delayed action and more time in which to spray, vaporize, and burn the fuel; hence lower cylinder pressures are possible, permitting the construction of lighter and cheaper engines which have nearly the same fuel economy as the true Diesels.

A sectionalized view of a Worthington two-cycle precombustion engine is shown in Fig. 261. The cylinders, whose axes are vertical, are cast in one piece including water jackets and liners. Crossheads and guides of the double-bored type are provided to prevent excessive wear of cylinder and piston.

Scavenging air is compressed in the enclosed spaces surrounding the crosshead guides, which spaces are entirely separated from the crankcase; thus the air is not contaminated by fumes from the crankcase. The air spaces and piston displacement are so proportioned that the air pressure at the end of the downward stroke of the piston is 3 or 4 lbs. per sq. in. above atmospheric. This pressure is sufficient for satisfactory scavenging and charging the cylinder.

In small engines which develop up to 60 or 75 horsepower per cylinder, no special provision for piston cooling is necessary, but for the larger sizes extra cooling must be provided by water or oil circulation. The lubricating oil, passed through a suitable cooler, is a satisfactory cooling agent in sizes up to about 150 horsepower per cylinder. But in larger engines, some of which develop up to 1500 horsepower per cylinder, water seems to be the only effective medium.

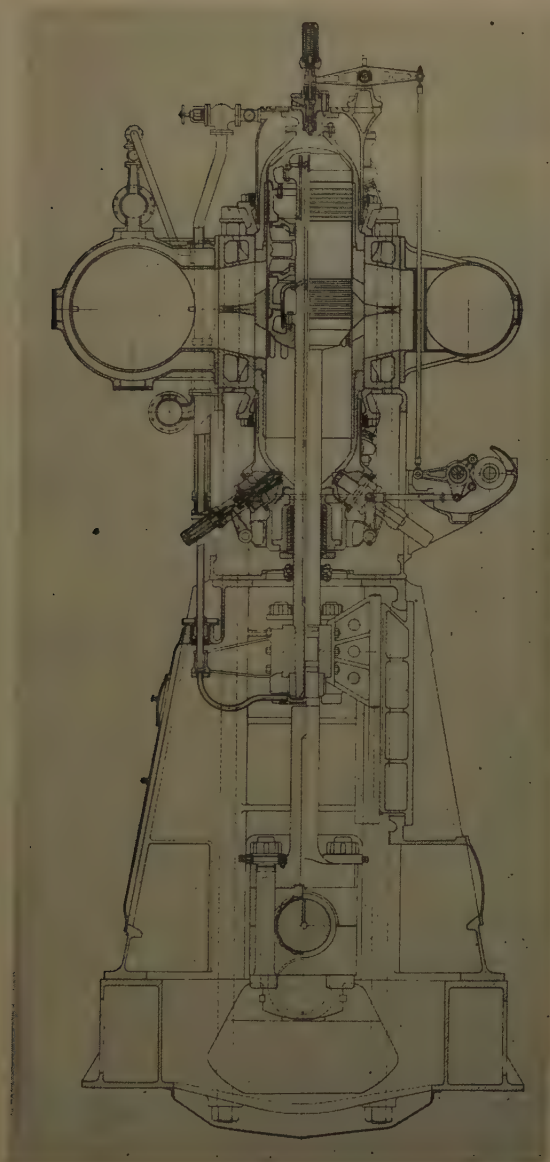
Worthington two-cycle precombustion engines are built for

both land and marine service. The marine engines are equipped with reversing gear to increase the maneuverability. Both types are built in sizes ranging from 55 to 540 B.hp. as indicated in Table 73.

TABLE 73
SPECIFICATIONS FOR WORTHINGTON DIESEL ENGINES

Rated D.hp. per Cylinder	Cylinder Dimensions Inches	Rated R.p.m.	Number of Cylinders	Per Cent Load Specific Fuel Rates		
				50	75	100
SINGLE-ACTING TWO-CYCLE SOLID-INJECTION PRECOMBUSTION TYPE						
18.6	7.25 × 10.00	450	3 or 4	0.65	0.54	0.52
30.0	10.25 × 10.50	375	2, 3, or 4	0.60	0.51	0.50
60.0	12.50 × 13.25	325	2, 3, or 4	0.56	0.50	0.49
90.0	15.25 × 16.00	275	3, 4, 5, or 6	0.49	0.48	0.46
FOUR-CYCLE SINGLE-ACTING AIR-INJECTION TYPE						
105	17 × 25	200	3, 4, 5, 6, or 8	0.48	0.43	0.42
120	17 × 25	225	3, 4, 5, 6, or 8	0.48	0.43	0.42

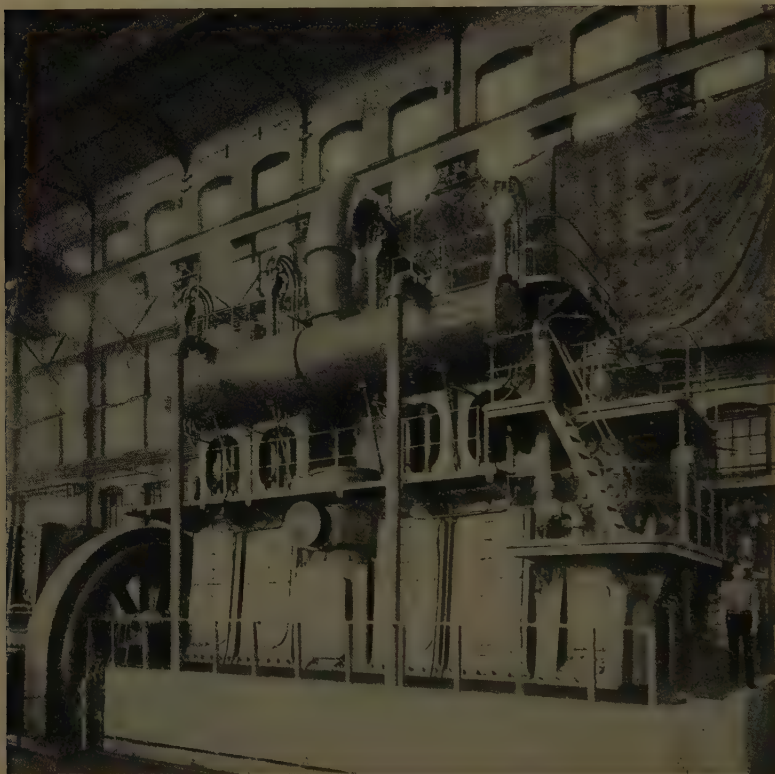
238. Worthington Two-Cycle Double-Acting Diesel Engine. The most rapid development in the Diesel-engine field of recent times, and one which no doubt will prove to be of great importance in power generation in medium and large units, is the double-acting design. In 1920 this design was considered to be in a highly experimental stage. Now six builders have large double-acting engines in marine service, and five others have about completed development work on double-acting units. The double-acting engine is especially well adapted for marine service because of its weight (200 lbs. per hp. or less). It is being readily adopted by ship-owners for cargo boats because it closely resembles the familiar steam engine in structure and operating characteristics. There are several factors which have retarded wide acceptance of the Diesel as a prime mover. It could not be built in large powers, 2000 or 3000 hp. units being the largest available up to about 1920. But the double-acting engine may



(Worthington Pump and Machinery Corporation)

FIG. 262.—DOUBLE-ACTING, TWO-CYCLE AIR-INJECTION DIESEL ENGINE

be made much larger. At present the largest Diesel in existence is a nine-cylinder, 15 000 B.hp., double-acting engine built for stand-by service in the municipal plant of Hamburg, Germany.¹ Double-acting units developing 25 000 B.hp. would seem to be entirely possible in the not-distant future. Another obstacle to



(Worthington Pump and Machinery Corporation)

FIG. 263.—28 × 40 DOUBLE-ACTING, TWO-CYCLE AIR-INJECTION
DIESEL ENGINE

more general adoption of Diesels has been their high first cost and high maintenance charges. The double-acting principle permits of unit-price reduction, and intensive development and improvement is rapidly reducing maintenance costs. As time passes, the large double-acting Diesels seem destined to play an increasingly

¹ For description and test results, see pp. 579 to 583 herein.

important rôle in power production, especially in the marine field.

Views of the Worthington, double-acting two-cycle air-injection Diesel are shown in Figs. 262 to 263. After four years of development work, the completion of the first Worthington double-acting engine was announced in 1924. This was the first double-acting Diesel brought out in this country. At that time four European firms, Burmeister and Wain, North British, Werkspoor, and M.A.N. (Manschinenfabrick Augsburg-Nürnberg) were already producing double-acting engines commercially.

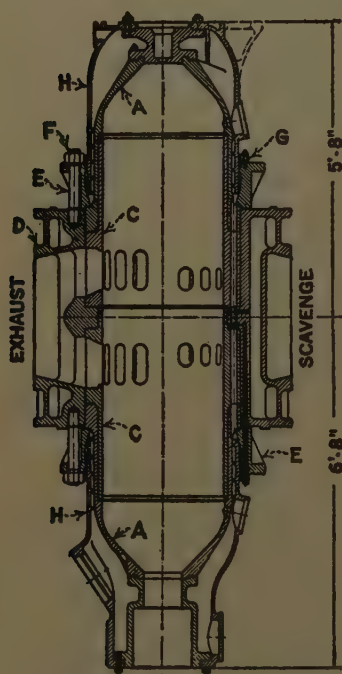
Certain problems have had to be solved in developing the double-acting Diesel. In the steam engine, the chief factor responsible for high economy was to prevent cylinder condensation by keeping the cylinder *hot enough*. In the internal combustion engine, the problem is to keep the cylinder *cool enough* to maintain the strength of the metal, and to prevent excessive stresses and unequal expansion due to high temperatures. The quantity of heat produced in a cylinder varies as the volume, which in turn varies as the cube of the cylinder diameter; while the surface for carrying away the heat varies only as the square of the diameter; hence in large cylinders the quantity of heat to be carried away is much greater per square foot of cooling surface. Furthermore, the walls for the large cylinders must be thicker to withstand the greater forces. The heat generated per unit of time in a two-cycle cylinder is greater than in a four-cycle for the same power. Hence the problem of cylinder and piston cooling is doubly difficult in two-cycle double-acting engines of large power.

A cylinder design which is entirely new was developed to solve the problems mentioned above. It consists of two single-acting cylinders placed opposite each other with their pistons on the same rod; each cylinder is provided with its own scavenging-air and exhaust ports. The division between the cylinders is made very short, and the two piston heads are extended into a single long trunk piston with semispherical-shaped ends as seen in Fig. 262.

The cylinder consists of two dome-shaped steel forgings *A* and a common central cast-iron base *D* containing the exhaust

and scavenging ports. The large end of each half-cylinder carries a substantial shoulder by which it is secured to the central base by means of clamping rings *E* and studs *F*. By this construction, the cylinder is fastened at its coldest zone, leaving the hotter ends free to expand. A light cast-iron jacket *H* is placed

around each cylinder. It is bolted to the end of the cylinder, and supported at the inner end by the clamping ring, against which a water-tight joint is made by the rubber packing-ring *G*. The forms of the top and bottom cylinders differ from each other somewhat. There is only one fuel valve, centrally located, in the top cylinder. In the bottom cylinder, provision must be made for the piston-rod packing and two fuel valves, which enter the cylinder just above the packing gland, and at an angle with the axis of the piston rod.



(Worthington Pump and Machinery Corporation)

FIG. 264.—CYLINDER FOR DOUBLE-ACTING TWO-CYCLE AIR-INJECTION DIESEL ENGINE

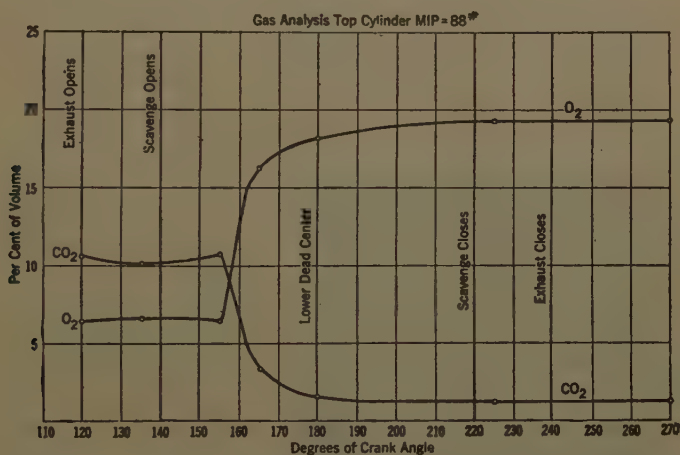
A cast-iron liner *C* (cast iron for the wearing surface being preferred to steel) is pressed into the forged-steel cylinders. The forged cylinders carry all the stress due to internal pressure; hence the cast-iron liners may be made rather thin to give the

greatest possible heat conductivity. The inner ends of both liners are enlarged to accommodate the cylinder ports. The inner ends of the liners do not butt solidly together, a small space being left to allow for expansion. The clamping-ring *E* holds the liners in place.

The combustion chamber approaches the ideal spherical form. It is free to expand in all directions, and has smooth surfaces

without corners or projections. This construction avoids severe local force concentrations due to heat strains.

The hollow piston heads are of forged steel, but the sleeves carrying the piston rings are of cast iron. The piston is built up of parts bolted together. This would seem to be a serious objection because of the danger of the parts working loose when the engine is in operation. The piston is water-cooled. The cooling water is conveyed to and from passages in the rod by means of telescoping tubes attached to the crosshead.



(Worthington Pump and Machinery Corporation)

FIG. 265.—ANALYSES OF CYLINDER GASES OF DOUBLE-ACTING TWO-CYCLE DIESEL ENGINE

A marine type crosshead, carrying a bracket to which the piston cooling-gear is connected, is provided to guide the piston.

Scavenging and recharging the cylinders with air are accomplished through central ports. In the older units, the scavenging air is supplied by two reciprocating compressor cylinders placed in tandem, and driven from a crank on an extension of the main shaft. In the latest units, the scavenging air is supplied by centrifugal compressors driven by electric motors. The perfection of the scavenging process was determined by analyzing the cylinder gases during the scavenging period when the engine was overloaded 10 per cent. The results of the analysis are shown by

Fig. 265. This analysis shows that the oxygen content of the cylinder reached a maximum of 19.5 per cent by volume. The content in the atmosphere is 20.9 per cent. Hence the scavenging efficiency is seen to be very satisfactory.

The high-pressure air for fuel spray is provided by a four-stage compressor. The first stage is double-acting, and the other three stages are single-acting.

Each fuel spray valve is operated by a separate cam, which is symmetrical, permitting it to rotate in either direction. Each valve lever oscillates about an eccentric on the fulcrum shaft, which can be rotated through a small angle by a hand lever, giving a reduced valve lift at slow speeds.

The engine is reversed by rotating the cam shaft through an angle of 34 degrees relative to the main shaft. The reversing



(Worthington Pump and Machinery Corporation)

FIG. 266.—REVERSING GEAR FOR DOUBLE-ACTING TWO-CYCLE DIESEL ENGINE

gear (Fig. 266) consists of a spur gear having helical teeth which engage an idling gear of some length. The idler drives a spur gear keyed to a horizontal lay shaft located in the bed plate on the engine front. The lay shaft can be shifted axially in its bearings just enough to cause it to rotate 34 degrees due to the helical

teeth. The lay shaft passes through a sleeve which rotates with it. The cam shaft is driven from a bevel gear bolted to the sleeve, and a vertical intermediate shaft. The cam shaft will, therefore, be displaced 34 degrees when the lay shaft slides from one extreme position to the other. The lay shaft is attached to a piston by means of two ball bearings. The piston travels in a cylinder, being held in one or the other extreme end of the cylinder by oil pressure. Engine reversal is accomplished by simply turning a four-way cock, permitting oil to escape from one end, and admit-

ting oil to the other end of the cylinder. During maneuvering tests on a 4-cylinder 2900 hp. engine, it was found that the time required to reverse the engine from full-speed-ahead to the first ignition in the astern direction varied from about 5 to 8 seconds.

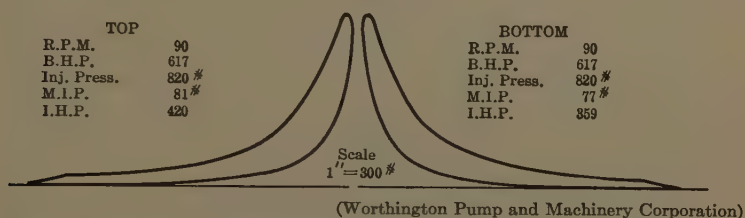


FIG. 267.—INDICATOR CARDS FROM DOUBLE-ACTING TWO-CYCLE DIESEL ENGINE

Indicator cards taken from an experimental one-cylinder double-acting 27×40 engine, developing 617 brake horsepower are shown in Fig. 267. The indicated pressures and horsepower are made somewhat smaller in the bottom cylinder in order to keep the load on the piston-rod packing within reasonable limits and to prevent overheating the piston rod.

239. Diesel-Engine Performance Results. Summarized results of official acceptance tests of the first two Worthington double-acting two-cycle Diesel engines which were built for the U. S. Shipping Board are given in Table 74. A summary of the official tests at different loads of two engines turned over to the Board in 1928 are presented in Table 75. It will be noted that the fuel-load mechanical efficiency and fuel rate are somewhat better for the latest engines.

In Table 77 are given tests result at various loads for one of three, two-cycle single-acting air-injection six-cylinder Nordberg engines installed in 1927 at the Panama Canal. The cylinders are 29×40 inches, and the engine speed is 125 r.p.m. The weight of the engine alone is 815 000 lbs. or about 218 lbs. per B.hp.

Table 78 contains a summary of the acceptance-test results for the Blohm and Voss nine-cylinder double-acting two-cycle Diesel engine purchased for stand-by service in the Neuhoof municipal power station of the city of Hamburg.

TABLE 74

SUMMARY OF TEST RESULTS FOR WORTHINGTON DOUBLE-ACTING TWO-CYCLE
DIESEL ENGINES BUILT FOR THE U. S. SHIPPING BOARD

ENGINE FOR	M.S. TAMPA	M.S. UNICOI	ENGINE FOR	M.S. TAMPA	M.S. UNICOI
Date of test.....	Feb. 2 to Mar. 4, 1926	May 5 and 6, 1927	<i>Temperatures, deg. F.</i>		
Bore in inches.....	28	28	Exhaust.....	548.1	513.3
Stroke in inches.....	40	40	Lubricating oil.....	96.0	89.8
Number of cylinders ..	4	4	Fuel oil.....	88.6	73.6
Duration of test, hours.	720	24	Engine room.....	67.9	54.8
Revolutions per minute	95.4	95.32	Atmosphere.....	29.3	
Brake horsepower	2926.71	2911.64	<i>Air temperatures, deg. F.</i>		
Indicated horsepower ..	4038.05	4094.90	1st stage cooler inlet..	142	146
Mean indicated pressure	89.47	90.85	1st stage cooler outlet..	43	79
Mean effective pressure	64.8	64.6	2d stage cooler inlet..	139	177
Mechanical efficiency,			2d stage cooler outlet..	43	79
per cent.....	72.48	71.1	3d stage cooler inlet..	138	180
Thermal efficiency, per			3d stage cooler outlet..	42	79
cent.....	29.08	30.84	4th stage cooler inlet..	234	269
Air compressor, I.hp...	187.3	180.65	4th stage cooler outlet..	77	108
Scavenging pump, I.hp..	182.2	187.85	<i>Pressures, lbs. per sq. in.</i>		
Fuel oil, lbs. per hour ..	1426.69	1340.96	Injection air.....	889	878.2
Fuel oil, lbs. per I.hp.			3d stage of compressor..	246	211
hour *.....	0.342	0.317	2d stage of compressor..	74	69
Fuel oil, lbs. per B.hp..			1st stage of compressor	24	19
hour *.....	0.473	0.446	Scavenging air.....	1.69	1.69
<i>Lubricating oil, gals. per</i>			Lubricating oil.....	12	12
<i>24 hours</i>			Cooling water.....	30.3	27.4
Power cylinders.....	3.19	5.25	Barometer (inches)...	29.06	
Stuffing boxes.....	3.92	2.75	<i>Fuel characteristics</i>		
Air compressor.....	0.57	0.75	Gravity, Be. at 60° F.	20.3°	21.3°
Scavenging pump.....	0.25	1.38	Viscosity, Saybolt....		
Hand feeds.....	0.28	0.26	Fuel at 77° F.....	67.2	62.0
Crankcase.....	8.79	7.07	Flash point, Pensky-		
			Martens, deg. F....	182	182
All purposes.....	17.00	17.46	Burning point, deg. F.	273	273
<i>Cooling water tempera-</i>			Sulphur content, per		
<i>tures, deg. F.</i>			cent.....	0.40	0.40
Inlet to engine.....	39.4	79.7	High heating value,		
Outlet from engine....	116.4	120.3	B.t.u. per pound ..	19 002	19 002
			Low heating value,		
			B.t.u. per pound...	17 917	17 917

* Corrected to the higher heating value of 18 500 B.t.u. per lb.

The engine is rated at 15 000 metric ¹ brake horsepower at 94 r.p.m. Air, at a pressure varying in accordance with the load from about 800 to 1000 lbs. per sq. in., is employed to spray the fuel into the cylinders. The air compressor is driven from an extension of the main shaft, but the rotary scavenging pumps (two in number) are driven by direct connected electric motors. The engine, without the electrical generator, weighs about 1230 tons or about 165 lbs. per horsepower.

The fuel economy and thermal efficiency attained are excellent for a two-cycle engine, nearly equaling the best performance of

¹ 1 metric hp. equals 0.986, English hp.

TABLE 75¹

SUMMARY OF ACCEPTANCE TESTS OF WORTHINGTON 3725 HORSEPOWER
(28 × 40) TWO-CYCLE DOUBLE-ACTING DIESEL ENGINE, DEC. 12
TO DEC. 19, 1928

Per cent of rated load	24.4	47.4	68.3	100.0	104.0
Duration of test	2 hrs.	4 hrs.	6 hr.	7 days	6 hrs.
R.p.m.	72.3	91.0	98.1	115.0	119.5
M.I.P. top end lbs. per sq. in.	48.5	66.8	73.6	87.3	84.2
M.I.P. bottom end lbs. per sq. in.	36.7	57.8	67.8	84.6	84.5
I.hp. top end	875	1512	1797	2498	2502
I.hp. bottom end	591	1175	1487	2174	2252
I.hp. total	1466	2687	3284	4672	4754
Brake horsepower	909	1758	2547	3723	3874
Mechanical efficiency per cent.*	62.0	65.4	77.5	79.7	81.5
Scavenging blower r.p.m.	2100	2300	2325	2490	2540
Scavenging blower d.hp.	97.4	131	140	171	182
Fuel consumption lbs. per hour	467	780	1063	1544	1584
Fuel rate, lbs. per I.hp. per hr. †	0.316	0.288	0.321	0.327	0.331
Fuel rate, lbs. per d.hp. per hr. †	0.509	0.440	0.414	0.411	0.406
Brake thermal efficiency per cent	26.4	30.4	32.4	32.5	33.0

* Does not include power to drive the scavenging blower.

† Corrected to 19 000 B.t.u. higher heating value.

Weight of engines	735 676 lbs.
Weight of 2-blowers	30 000 lbs.
Total weight	765 676 lbs.
Weight per D.hp. (rated)	206 lbs.

TABLE 76

TEST RESULTS OF ATLAS-IMPERIAL SIX-CYLINDER SOLID-INJECTION FOUR-
CYCLE SINGLE-ACTING DIESEL ENGINE

Per cent of rated load	100
R.p.m.	277
Cylinder diameter, inches	11.5
Piston stroke, inches	15
Pressure of fuel at injection, lbs. per sq in.	4000
I.hp.	392.9
D.hp.	300.0
Mechanical efficiency, per cent	76.5
Specific fuel rate, lbs. per D.hp. per hr.	0.393
* Brake thermal efficiency per cent.	35.1

* Based on the higher heating value of 18 500 B.t.u. per lb.

¹ Data in Tables 74 and 75 was supplied by the Worthington Pump and Machinery Corporation.

TABLE 77¹

TEST RESULTS OF A NORDBERG SIX-CYLINDER (29" × 40") TWO-CYCLE
SINGLE-ACTING DIESEL ENGINE RATED AT 3750 B.H.P. AT 125 R.P.M.

Per cent of rated load	25.15	50.4	75.4	100.1	110.2
R.p.m.	125.6	125.1	125.2	124.7	125.0
Pressure of injection air, lbs. per sq. in. gage.	823	974	1030	1070	1075
Pressure scavenging air, lbs. per sq. in. gage.	5.16	4.90	5.00	5.00	5.10
Temperature of exhaust gases, deg. F.	315	400	510	640	690
Mean indicated pressure, lbs. per sq. in.	37.28	57.10	71.45	89.57	97.40
I.hp. power cylinders	2063	3150	3945	4925	5360
I.hp. air compressor	171.6	212.4	236.3	250.2	249.9
I.hp. scavenging pump	387.5	385.0	376.2	386.0	395.0
Developed horsepower	943	1886	2823	3759	4143
Mechanical efficiency, per cent. .	45.8	59.9	71.6	76.3	77.2
Fuel oil, total lbs. per hr.	584.0	871.0	1218.8	1585.6	1728.0
Specific fuel rate, lbs. per D.hp. per hr.	0.618	0.462	0.431	0.422	0.417
Higher heating value B.t.u. per lb.	18 738	18 738	18 738	18 738	18 738
Brake thermal efficiency, per cent.	21.9	29.4	31.5	32.2	32.6

the best four-cycle engines. However, in making comparisons between American and European engines, it should not be forgotten that the thermal efficiencies of American engines are apparently low because they are usually based upon the *higher heating value* of the fuel, and thermal efficiencies of European engines are based upon the *lower heating value*.

The distribution of the heat energy for the Blohm and Voss engine at full load is shown in Table 79. These results are typical of all air-injection Diesel engines.

The results recorded in Table 80 typify the best that can be expected of large or small Diesel engines. The thermal efficiencies appear to be unduly high because they are based on the lower heating value of the fuel.

¹ See Power Plant Engineering, May 15, 1927, p. 563.

TABLE 78¹

SUMMARY OF TESTS OF BLOHM AND VOSS NINE-CYLINDER TWO-CYCLE
DOUBLE-ACTING DIESEL ENGINE RATED AT 14 795 B.H.P. AT 94
R.P.M.—CYLINDER BORE 860 MM. (33.86")—STROKE
1500 MM. (59.055")

Per cent of rated load	25.2	51.6	75.2	97.6
R.p.m.	94	94	94	94
Pressure of injection air, lbs. per sq. in. g .	782	853	924	1010
Pressure of scavenging air, lbs. per sq. in. gage.	1.720	1.684	1.733	1.75
Temperature of exhaust gases, deg. F. . .	221	322	428	545
Mean indicated pressure, lbs. per sq. in. .	29.29	48.06	64.84	80.05
I.hp. (550 ft. lbs. per sec.) exclusive of scavenging-blower power.	6314	10 317	13 925	17 210
D.hp. (550 ft. lbs. per sec.)	3728	7634	11 125	14 440
h.p. to drive scavenging blower	764	745	764	767
Mechanical eff. (incl. scavenging blower). .	52.7	69.0	75.7	80.3
Fuel oil total (exclusive of scavenging blower), lbs. per hr.	1731	2864	4101	5435
Specific fuel rate (including scavenging blower), lbs. per D.hp. per hr.	0.584	0.416	0.396	0.398
* Brake thermal eff. (incl. scavenging blower per cent)	24.2	34.0	35.7	35.5

* Based on the lower heating value of 18 018 B.t.u. per lb.

The Doxford engine (test results in Table 81) is made after the design proposed by Professor Junker of Germany. The vertical cylinders are without heads. The explosion takes place between two opposed pistons which move in opposite directions from the middle of the cylinder; hence there are no cylinder heads. The lower piston is attached to the main crank through a crosshead.

TABLE 79

THERMAL BALANCE OF 14 795 HP. DIESEL ENGINE AT FULL LOAD

Heat supplied in full	100.0 per cent
Heat carried away in cooling water.	26.1 " "
Heat carried away in exhaust and radiation	29.3 " "
Heat lost in friction	4.2 " "
Heat equivalent of compressor work.	3.1 " "
Heat equivalent of scavenging blower.	1.8 " "
Heat equivalent of useful work	35.5 " "

¹ See Engineering (London), July 23, 1926, p. 119, and Oct. 29, 1926, p. 555;
Zeitschrift des Vereines deutscher Ingenieure, Oct. 23, 1926, p. 1409.

TABLE 80¹

TEST RESULTS OF A KRUPP SINGLE-ACTING FOUR-CYCLE AIRLESS-INJECTION
DIESEL ENGINE DEVELOPING 513 HP. CYLINDER DIMENSIONS, BORE
460 MM. (18.11"); STROKE 630 MM. (24.81")

Per cent of rated load	25.7	50.0	74.5	100.5	111.5
R.p.m.	221.0	220.0	218.6	215.6	213.5
Mean indicated pressure, lbs. per sq. in.	36.6	54.8	73.6	91.8	103.3
Indicated hp. (550 ft. lbs. per sec.) .	245.8	375.0	502.0	619.0	686.5
Developed hp. (550 ft. lbs. per sec.) .	131.9	256.5	380.0	516.0	572.0
Mechanical efficiency, per cent. . .	53.7	68.0	76.2	83.5	83.4
Total weight of fuel per hr. lbs. . . .	66.4	106.5	144.8	192.2	215.5
Specific fuel rate lbs. per D.hp. per hr.	0.504	0.415	0.381	0.372	0.376
* Thermal efficiency based on D.hp. per cent	27.7	33.6	36.6	37.5	37.1

* Based on the lower heating value of 18 280 B.t.u. per lb.

TABLE 81²

TEST RESULTS OF A DOXFORD MARINE FOUR-CYLINDER (22.83" × 45.67")
TWO-CYCLE SINGLE-ACTING SOLID-INJECTION DIESEL ENGINE RATED
AT 2900 B.H.P. AT 87 R.P.M.

Per cent rated load	12.75	35.6	64.2	100.3	125.6
R.p.m.	43.8	61.3	74.5	87.4	94.8
Temperature of exhaust gases, deg. F.		270	419	593	636
Mean indicated pressure, lbs. pr. sq. in.	32	54	75	101	115
Indicated horsepower (correct- ed)	530	1230	2150	3330	4160
Developed horsepower	370	1020	1860	2910	3640
Injection-fuel pressure, lbs. pr. sq. in.	7 700	4 700	7 000	9 200	10 200
I.hp. scavenging pump	20	56	103	178	227
Mechanical efficiency, per cent. .	68.9	83.0	86.7	87.5	87.4
Total fuel corrected, lbs. per hr. .	174	420	750	1212	1536
Specific fuel rate, lbs. per S.hp. .	0.471	0.412	0.403	0.416	0.424
Higher heating value of fuel, B.t.u. per lb.	18 220	18 270	18 220	18 220	18 220
Brake thermal efficiency, per cent	29.7	33.9	34.6	33.6	33.1

¹ See Zeitschrift des Vereines deutscher Ingenieure, Nov. 27, 1926, pp. 1597-1604.

² Proceedings Institution of Mechanical Engineers, Part 1 (1926), pp. 99-213, for description and test results.

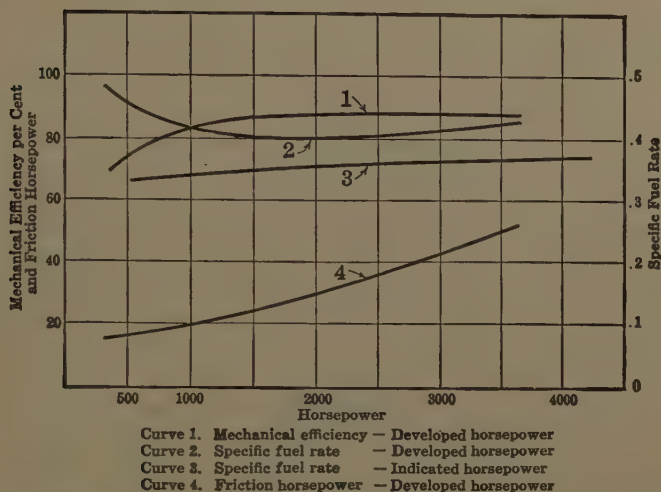


FIG. 268.—PERFORMANCE CURVES OF 2900 HP. DOXFORD MARINE DIESEL ENGINE

The performance curves (Fig. 268) for the Doxford 2900-horsepower engine are more or less typical for all Diesel engines which are operated at reduced speed at the partial loads, which is the procedure followed in all marine engine operation.

“The top piston is attached to a T-shaped yoke which is connected through a pair of long pull rods to two crossheads on either side of the cylinders. These crossheads are connected by rods to two cranks 180 degrees out of phase with the main crank. Therefore there are three cranks per cylinder, and except for instantaneous differences of resultant thrust due to the obliquities of the connecting rods, there is no combustion load on the main bearings.”¹

The weight of the engine per D.hp. is rather high because of the low rotative speed. The weight including thrust block, all pipe and fittings, auxiliary fuel pump, and flywheel, is 740 000 lbs., or 255 lbs. per D.hp.

The curves of Fig. 269 are Willans lines, but they do not turn out to be straight lines (nor nearly so) as in the cases of steam engines and turbines. However, in the case of Otto-cycle engines

¹ Quotation from Proceedings, Institution of Mechanical Engineers, Part 1, 1926.

using gaseous fuels, the Willans line is practically straight. The curves are plotted from test results of engines as listed below.

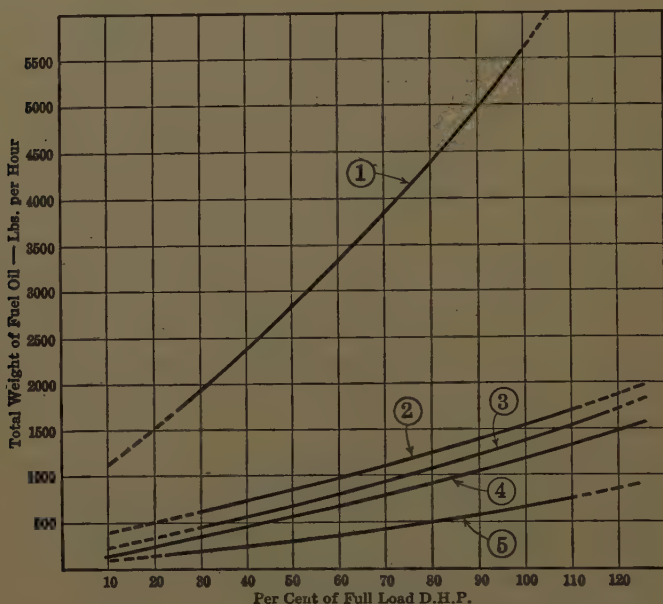


FIG. 269.—WILLANS LINES FOR DIESEL ENGINES

- Curve 1, Blohm and Voss, 15 000 metric D.hp., 9-cylinder, double-acting Diesel (Table 78).
- Curve 2, Nordberg, 3750 D.hp., 2-cycle, single-acting, Diesel engine (Table 77).
- Curve 3, Worthington, 2900 D.hp., 2-cycle, double-acting, Diesel engine (Tables 74 and 75).
- Curve 4, Doxford, 2900 D.hp., marine, 2-cycle, solid-injection, Diesel engine (Table 81).
- Curve 5, Nobel, 1600 D.hp. 2-cycle, single-acting, Diesel engine (Table 82).

The range of power, speed, and weight per horsepower of internal-combustion engines are given in Table 83. The power of engines in use ranges all the way from one horsepower or less to 15 000 horsepower per engine. The weight range is rather extreme, varying all the way from 0.5 pound to 500 pounds per horsepower. In general the engines operating at the higher speeds are the lighter. The cost of engines per horsepower

TABLE 82¹

TEST RESULTS OF A NOBEL MARINE FOUR-CYLINDER (675 MM. $\times$ 920 MM.
EQUAL TO 26.58" $\times$ 36.21") TWO-CYCLE SINGLE-ACTING AIR-IN-
JECTION DIESEL ENGINE RATED AT 1600 HP. AT 106 R.P.M.

Per cent of rated load.....	23.15	49.9	76.0	100.8	108.1
R.p.m.....	45.2	87.3	85.0	107	105.1
Temperature of exhaust, deg. F.....	277	370	527	599	633
Mean indicated pressure, lbs. per sq. in.....	54.8	63.4	90.7	92.1	98.7
Indicated horsepower (550 ft. lbs. pr. sec.).....	489.5	1120	1561	1997	2100
Developed horsepower (550 ft. lbs. per sec.).....	370	799	1218	1611	1730
I.hp. of air compressor (550 ft. lbs. pr. sec.).....	50.2	79.4	93.1	110.2	113.1
I.hp. of scavenging pump, (550 ft. lbs. per. sec.).....	11.5	52.1	51.7	75.0	71.3
Mechanical efficiency, per cent.....	75.6	71.3	78.1	80.7	82.3
Fuel, total lbs. per hr.....	156.1	329	496	661.5	706.5
Specific fuel rate, lbs. per D.hp.....	0.422	0.412	0.4075	0.410	0.4085
* Brake thermal efficiency, per cent.....	33.65	34.55	34.95	34.75	34.90

* Based on lower heating value of 17 900 B.t.u. per lb. of oil.

varies somewhat in accordance with the weight, except in the extreme case of airplane engines, hence there is a continual effort made to reduce the unit weight by increasing the economical speed at which the engines may be operated.

240. Combustion and Exhaust-Gas Analysis. Analysis of the exhaust gases of an internal-combustion engine is the only available method of determining the degree of perfection attained in the combustion. This analysis, together with the ultimate chemical analysis of the fuel, furnishes the means by which the air theoretically required, the amount of air actually supplied, and whether there is adequate mixing of the air and fuel, may be determined.

¹ See The Engineer, Jan. 27, 1922, p. 91, and Feb. 3, 1922, p. 126.

TABLE 83¹

POWER, SPEED, AND WEIGHT OF INTERNAL COMBUSTION ENGINES

CLASS *	FUEL	POWER RANGE D.H.P.	SPEED RANGE R.P.M.	WEIGHT	
				Range, Lbs. per D.hp.	Aver.
Airplane	Gasoline	100-1500	1200-2750	0.5-4	2.5
Automobile	Gasoline	20-265	1500-7000	8-20	12
Tractor	Gasoline	25-200	600-1500	20-75	50
Industrial	Gasoline	1-600	200-2000	10-200	...
Industrial	Gas	2-5000	75-400	40-400	...
Diesel truck ²	Diesel oil	50-100	800-1400	20-30	25
Diesel, 4C, SA, U- boat	Special oil	200-3000	300-750	50-100	70
Diesel, 4C, SA	Heavy oil	100-5000	85-750	100-500	300
Diesel 4C, DA	Heavy oil	1000-10 000	100-150	180-250	225
Diesel 2C, SA	Heavy oil	50-4000	100-450	100-300	250
Diesel 2C, DA	Heavy oil	2000-15 000	75-125	160-250	200

¹ 4C = four-cycle; 2C = two-cycle; SA = single-acting; DA = double-acting.

Petroleum and all the fractions obtained from it have practically the same composition as shown by the ultimate analysis. The average percentages of the principal components of different petroleum products are stated in Table 84. For gaseous fuels, the variations in the percentages by weight are much greater, as may be seen by reference to Table 28, page 195, and by solving Problem 16, page 214 herein.

TABLE 84

ANALYSES OF PETROLEUM PRODUCTS
(Percentages by Weight)

ELEMENTS	GASOLINE	KEROSENE	CALIFORNIA FUEL OIL	MEXICAN FUEL OIL
C	83.5	84.5	85.5	82.8
H	15.5	14.5	12.0	11.2
N, O, and S	1.0	1.0	2.5	6.0
Total	100.0	100.0	100.0	100.0

¹ See Zeitschrift des Vereines deutscher Ingenieure, Aug. 7, 1926, p. 1064.² See Power, Mar. 27, 1928, p. 556.

A most convenient way to check combustion efficiency in an engine is to prepare a chart from computations based upon the analysis of the fuel under consideration similar to Fig. 270.¹ By making a partial analysis of the exhaust gases from the engine using the fuel for which the chart was prepared, reference to the chart will then show the air-fuel ratio at a glance. The chart may be used very advantageously also to check up on the combustion in a boiler furnace.

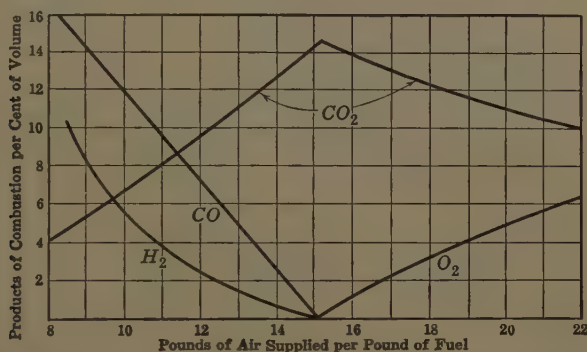


FIG. 270.—EXHAUST-GAS ANALYSES FOR GASOLINE (COMPUTED)

Typical exhaust gas analyses for internal-combustion engines are presented in Tables 85 and 86.

TABLE 85
ANALYSES OF EXHAUST GASES FROM GASOLINE AUTO ENGINES
(Percentages by Volume)

ANALYSIS NO.	1	2	3	4	5	6	7
Type of car.....	5-pass.	5-pass.	7-pass.	7-pass.	2-pass.	7-pass.	3½ ton truck
No. of cylinders....	4	4	6	6	4		
Car speed, miles per hr.....	20	25	15	12	38	15	10
Carbon dioxide.....	10.5	12.0	8.4	9.5	10.8	7.5	12.9
Oxygen.....	1.4	3.7	1.3	0.7	4.7	2.0	0.3
Carbon monoxide....	3.0	0.0	8.1	6.4	0.0	9.3	1.9
Methane.....						1.4	0.8
Hydrogen.....						4.0	0.4
Air—gasoline ratio..	12.7	18.5	11.1	11.9	20.5	10.7	13.9

Analysis No. 1, Table 85, was made on gases taken from a 4-cylinder 5-passenger car, equipped with a special carburetor, before it had gone one mile from a cold start.

¹ See Sec. Lieut. R. W. Fenning, *The Composition of the Exhaust from Liquid-Fuel Engines*, Proceedings, Institution of Mechanical Engineers, Jan.-May, 1916, pp. 185-236. For a better treatment and a more complete chart see Prof. E. H. Lockwood in S.A.E. Journal, Nov. 1927, p. 571, and Sept. 1928, p. 315.

- Analysis No. 2 was made on a sample of gas taken from same car as No. 1 analysis, no changes were made in carburetor adjustment, but the car was run about six miles to warm up.
- Analysis No. 3 was made on a sample taken from a 6-cylinder 7-passenger car equipped with a new standard carburetor. The car was driven about one mile from a cold start when the sample was taken.
- Analysis No. 4 is for the same car as No. 3. The engine was idled about fifteen minutes to warm up just before taking the sample. When the sample was taken, the car was climbing a 10 per cent grade with throttle about three-quarters open.
- Analysis No. 5 was made on gas taken from a 2-passenger 4-cylinder roadster soon after an "expert" mechanic had adjusted the carburetor to give what appeared to be the best all-round results.
- Analysis Nos. 6 and 7 were made by engineers of the U. S. Bureau of Mines on vehicles in exactly the same condition as furnished by the owners from whom they were borrowed for the purpose. The results are taken from a special report of the *Bureau of Mines serial No. 2225*, dated March, 1921.

TABLE 86
ANALYSES OF EXHAUST GASES FROM DIESEL ENGINES
(Percentages by Volume)

ANALYSIS NO.	1	2	3
Rated power, B.hp.....	2900	2900	1600
R.p.m. for test.....	96.4	87.4	107
Carbon dioxide, per cent.....	4.0	5.20	4.75
Oxygen, per cent.....	15.2	13.98	14.40
Carbon monoxide, per cent.....	trace	0.3	0.10
Sulphur dioxide, per cent.....	1.4		
Nitrogen, per cent.....	76.6	80.79	80.75
Water vapor, per cent.....	2.7		

Representative analyses of exhaust gases from Diesel engines are given in Table 86. It will be noted that the percentage of CO₂ is small and the excess oxygen is large in every instance. This is the usual condition in all Diesels. It is due to the fact that it is very difficult to spray the fuel into all the space of the combustion chamber; hence a large excess of air is required to prevent the formation of CO and smoke.

- Analysis No. 1 was made on gases from the Worthington 2900 hp. double-acting two-cycle marine engine, summary of test results for which is given in Table 74. The gas sample was taken while the engine was developing full load.
- Analysis No. 2 is for the gases taken from the Doxford 2900 horsepower engine, test results from which are given in Table 81.
- Analysis No. 3 is for the gases from the Nobel 1600 horsepower engine at full load. Test results for this engine are shown in Table 82.

PROBLEMS

1. The clearance volume of an Otto engine is 40 per cent. Compute the air-standard efficiency based on the value of $k = 1.4$.

2. Repeat Problem 1 when the clearance volume is 30 per cent.

3. Repeat Problems 1 and 2 using $k = 1.25$.

4. Compute the air-standard efficiency for the Chrysler Model 72 engine described on page 529.

5. The clearance volume of a Diesel engine is 7 per cent. Fuel injection and combustion for full load cease when the piston has completed 7 per cent of the power stroke. Compute the air-standard efficiency ($k = 1.4$).

6. Repeat Problem 5 when $k = 1.3$.

7. Compute the air-standard efficiency for the following specifications from a four-cycle dual-combustion engine: Pressure in the cylinder at the beginning of compression is 14.0 lbs. per sq. in. abs.; pressure at the beginning of constant volume combustion is 350 lbs. per sq. in. abs.; pressure during constant pressure combustion is 550 lbs. per sq. in. abs.; full-load combustion ceases when the piston has completed 5 per cent of its stroke.

8. Repeat for Problem 7 when the exponent for both the compression and expansion curves equals 1.30.

9. Compute the average indicated mean effective pressure for the engine from the test data of Table 76, page 581.

10. Compute the average brake mean effective pressure for the test data of Table 76.

11. Determine suitable cylinder dimensions and speed for a single-acting four-cycle four-cylinder Diesel engine to develop 1000 brake horsepower. The piston speed is to be about 900 feet per minute.

12. A gas engine operating on natural gas develops 500 B.hp., giving a brake thermal efficiency of 32 per cent, and the mechanical efficiency is 83 per cent. The higher heating value of the gas is 950 B.t.u. per cu. ft. Compute the cu. ft. of gas consumed per hour. Also compute the indicated thermal efficiency.

13. The following data is taken from a 12-cylinder Liberty airplane engine. Bore of cylinders, 5 ins.; stroke, 7 ins.; compression ratio, 5.4; higher heating value of the gasoline 20 300 B.t.u. per lb.; B.hp. at 1800 r.p.m., 410; fuel consumption, 0.52 lbs. per B.hp. per hr.; mechanical efficiency, 0.86.

Compute the air-standard efficiency; the brake thermal efficiency; the brake mean effective pressure; the indicated mean effective pressure; and the efficiency ratio referred to the brake horsepower.

14. A 2500-kw. Diesel-electric unit delivers 550 kw. hrs. per bbl. (42 gals.) of fuel oil whose gravity is 18° Beaumé. The higher heating value is 18 600 B.t.u. per lb.; mechanical efficiency of the engine is 0.80; the efficiency of the generator is 0.94. Compute the thermal efficiency of the entire unit; the brake thermal efficiency of the engine; and the fuel rate in lbs. per B.hp. per hr.

15. Determine suitable cylinder dimensions for a four-cylinder single-acting four-cycle Diesel engine for the conditions of Problem 14 having a piston speed between 850 and 950 feet per minute.

16. Determine suitable cylinder dimensions for the unit of Problem 11

for a four-cylinder double-acting two-cycle Diesel engine having a piston speed of 850 to 950 feet per minute.

17. Assuming the higher and lower heating values of gasoline to be 20 500 and 18 600 B.t.u. per lb. respectively, compute the brake thermal efficiency at 2000 r.p.m. for the Chrysler motor for both heating values (see curves of Fig. 233, page 531).

18. Compute theoretical exhaust-gas analyses for the gasoline whose analysis is given in Table 84, page 588, and plot curves similar to those of Fig. 270. Assume in every case that the hydrogen secures its oxygen first; then the carbon burns, first to CO and then to CO_2 , if there is sufficient oxygen.

19. Assuming that the analysis of California fuel oil, given in Table 84, is applicable, (a) compute the higher heating value of the fuel oil used in tests of the Krupp engine (Table 80, p. 584). (b) Compute the thermal efficiencies based upon the computed higher heating values for the various loads.

20. From the data given in Table 80 determine the number of engine cylinders.

21. Determine the mean effective pressure for the Whirlwind engine, p. 532, for 220 B.hp. at 1800 r.p.m. if the mechanical efficiency is 0.85.

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